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THE

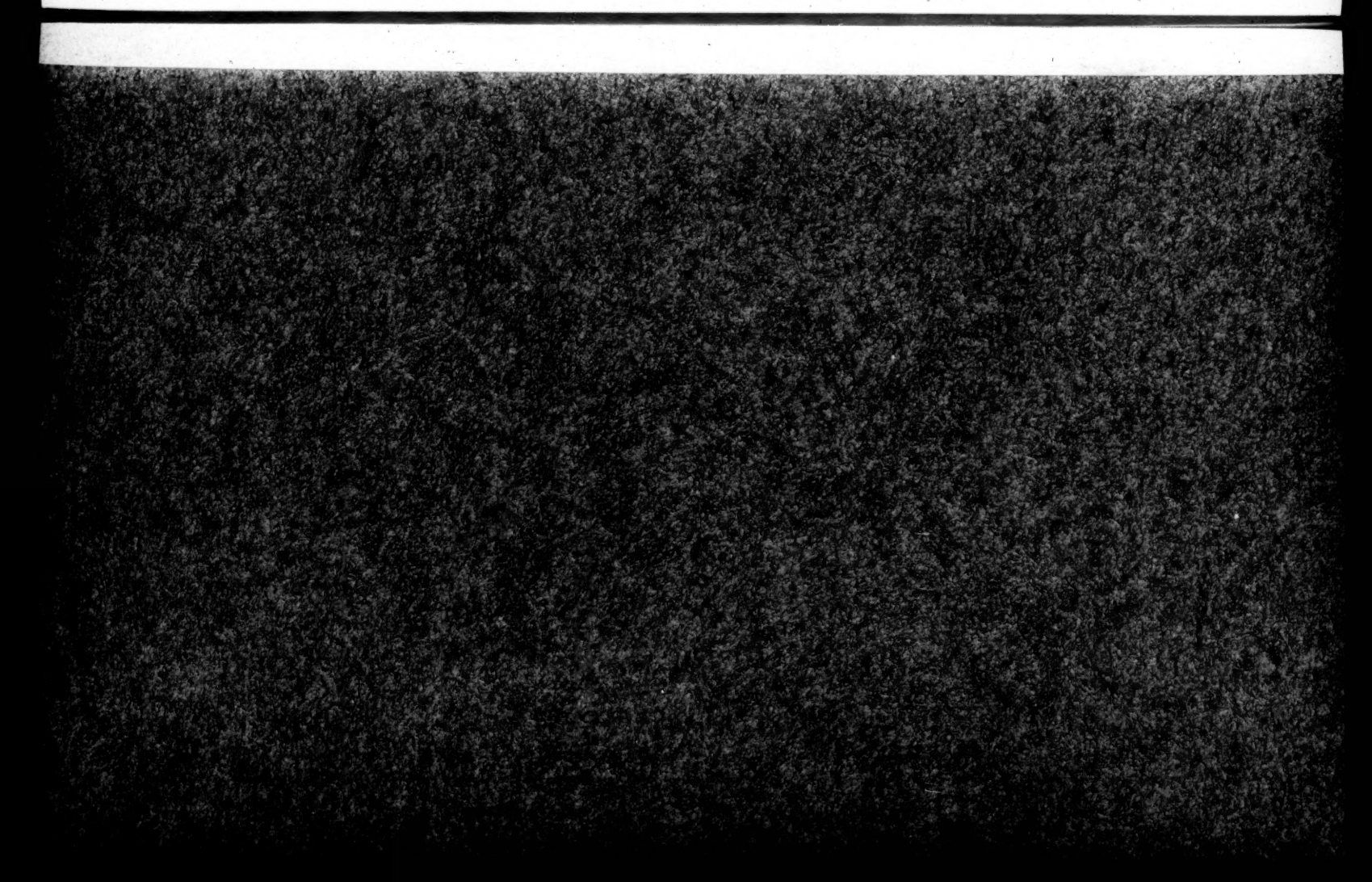
JOURNAL

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

CONTAINING
THE PROCEEDINGS

MID-OCTOBER 1909

MEETINGS OF THE SOCIETY: ST. LOUIS, OCTOBER 16; BOSTON, OCTOBER 20; NEW YORK, NOVEMBER 9; ANNUAL MEETING, NEW YORK, DECEMBER 7-10



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The Society as a body is not responsible for the statements of facts or opinions advanced in papers or discussions. C 55

HISTORY OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Preliminary Report of the Committee on Society History

Chapter X—*Continued*

CONTINUED GROWTH OF THE SOCIETY

The summer meeting of 1895 was held at Detroit, in June, and, as usual with the gatherings in the summer, the business transacted was kept at a minimum, being limited mainly to the necessary action upon candidates for membership, and similar routine business.

260 Mention has been made of the manner in which the general industrial situation is reflected in the meetings of the Society, and this fact was emphasized by the number and variety of papers presented at the Detroit meeting, and the bearing which they have had upon subsequent developments. Among the most important papers, viewed in this light, was undoubtedly that of F. W. Taylor, entitled A Piece-Rate System, introducing for discussion the differential piece-rate system afterwards more fully developed by the author, as well as the important idea of analyzing shop operations into their elementary details and determining the time required for these, so that their combinations cover the questions arising for the fixing of rates on various kinds of work. The methods employed for determining the laws governing the operations of metal-cutting, as laid down in this paper, have since been far more fully developed, and their investigation had doubtless much to do with the introduction of modern rapid-cutting tool steels. The discussion which this paper by Mr. Taylor elicited, showed very clearly the interest which the members of the

Under the direction of the Council the Committee on Society History has arranged to present the results of its investigations to the members of the Society.

The Preliminary Report will appear in The Journal of the Society from month to month, and thus enable the matter to be open to comment during its completion. It is especially desired that any member who may be in the possession of facts or information bearing upon the various points as they are thus made public will communicate with the committee, in order that the final and completed report may have the advantage of the collaboration of the membership at large.

Society took in matters relating to works management, an interest which has since become most widely extended and practically applied.

261 Another important contribution to engineering practice, presented at the Detroit meeting, resulted from the report of the committee on standard tests and methods of testing materials, presented by G. C. Henning, and supplemented by a report upon the transverse strength of cast iron by W. J. Keep, whose previous paper upon the strength of cast iron has already been mentioned. The exhaustive tests made by Mr. Keep upon the properties of cast iron, and the practical nature of his work and the conclusions drawn by him, render his contributions to the subject most valuable and reliable. This paper was followed by another, presenting to the Society and to the engineering profession Mr. Keep's study of "cooling curves," these curves showing the nature and extent of the shrinkage of the metal while cooling, and enabling information as to the strength and constitution of the casting to be derived.

262 The social features of the Detroit meeting were of an especially agreeable nature, including an excursion on the lake steamer City of Cleveland, a reception at the Detroit Club, and numerous visits to important industrial establishments.

263 Just prior to the Detroit meeting the Society experienced another loss from the ranks of its past-presidents, when Eckley B. Coxe, who had filled the presidential office for the two terms of 1893 and 1894, passed away on May 13, 1895. This loss was soon followed by another, most unexpected and sad, the death of the president of the Society, E. F. C. Davis, who was killed by a fall from his horse on August 6, 1895.

264 At the opening of the annual meeting in New York the first business was the announcement by Mr. J. F. Holloway of the action of the Council in the appointment, under the rules, of the senior vice-president, Charles E. Billings of Hartford, Conn., as president to fill the unexpired term of Mr. Davis. The address of Mr. Billings, discussing modern improvements in the drop press, formed another of the admirable monographs resulting from the adoption of the plan of selecting subjects intimately connected with the professional and practical work of the speaker. The tellers of election at this meeting announced the election of John Fritz to the office of president for the ensuing year. At this meeting there was presented a progress report of a committee on fireproof tests, a work undertaken in connection with similar committees representing the Fire Underwriters of New York and the Architectural League of New York.

265 At an example of the position which the Society had acquired as an organization recognized as a conservator of relics and documents relating to the historical side of engineering work, mention may be made of the fact that it was selected to receive the original autograph drawing of the steamboat Fulton, the first steamboat built for service on Long Island Sound. The drawing bears the date of March 1, 1813, as well as Fulton's signature. This drawing, formerly belonging to the late George L. Schuyler, of New York City, was presented to the Society by his daughter, Miss Louisa Lee Schuyler. The Society had already received, as a gift from Dr. Thomas Egleston, the handsome mahogany dining table formerly belonging to Fulton, so that it was already well on the way to become the recognized custodian of such relics.

266 At the same meeting the attention of the members was called to an excellent oil portrait of Captain John Ericsson, the work of the Swedish artist Ballin, a picture which had formerly belonged to Captain Ericsson, but after his death found its way into the hands of a dealer, from whom it was purchased by the Secretary of the Society.

267 The papers of the New York meeting of 1895 included a number devoted to the subject of power-plant design, including both steam and water power. Among the latter, attention may be called to the exhaustive paper upon water power by the veteran hydraulic engineer, Samuel Webber, this being a very full historical review of the development of the hydraulic turbine in America, especially in connection with the use of the power of the Merrimac river, and other power sources in New England. A paper by Samuel McElroy, upon the development of the water power of Caratunk Falls, in Maine, furnished an admirable pendant to the contribution of Mr. Webber, especially as showing the manner in which a hydraulic-power site may be investigated prior to the design of the plant, taking into account the experience gained in previous installation.s

268 Other papers at this meeting related to the use of the throttling calorimeter, for the purpose of determining the amount of moisture in steam; an important study, by Dr. Charles E. Emery, upon tests of steam boilers with different kinds of coal; and a discussion, by Prof. John H. Barr, of the proportions of high-speed engines. It is interesting to note that at this meeting the steam turbine began to receive serious discussion, a report of a test upon a small machine of the De-Laval type, by Professor Goss, attracting attention.

269 In accordance with arrangements made at the time of the Detroit meeting, the summer convention of 1896 was held at St. Louis,

Mo., in May. This meeting was devoted very largely to the discussion of matters relating to steam power plants. Dr. R. H. Thurston contributed a very complete paper upon superheated steam, while methods of determining the presence of moisture in coal received attention; and there was discussion of the performance of mechanical stokers. A paper by L. R. Alberger, upon what has since been called a "cooling tower" for enabling the use of the same condensing-water repeatedly, may be referred to as early mention of what has since become a standard piece of apparatus, this being but one example of the manner in which the meetings of the Society have served to bring into notice devices which have subsequently taken important places in commercial engineering work.

STRESSES IN REINFORCED-CONCRETE BEAMS

COMPARISON OF EXPERIMENTAL RESULTS WITH RESULTS OBTAINED FROM THE USE OF THREE THEORIES OF DISTRIBUTION OF STRESSES

By Prof. Gaetano Lanza, Boston, Mass.
Member of the Society

Lawrence S. Smith,[1] Boston, Mass.
Non-Member.

Many experiments have been performed on the breaking strength of reinforced-concrete beams, and in the course of them many observations have been made to determine quantitatively some of the phenomena attendant upon the application of the breaking load, and also upon that of smaller loads. Nevertheless it is well known that the observations made thus far are not sufficient to furnish the means for determining the actual distribution of the stresses, and hence for the deduction of reliable formulae for the computation of the direct stresses, shearing stresses, diagonal stresses, deflections, position of the neutral axis, etc., under a given load.

2 The test of the validity of such formulae should be their agreement with the results of experiments when the loads employed are about one-fourth or one-third the ultimate loads, because, when the loads are greater, the ratio of stress to strain varies very considerably for the different fibres, while for loads smaller than one-fourth of the ultimate, unknown initial stresses are liable to exert so great an influence as to interfere with the deductions.

3 The object of this paper is to make a comparison of (*a*) the position of the neutral axis, (*b*) the stress in the steel, (*c*) the stresses in the concrete, and (*d*) the deflection, as determined by experiment, with the same quantities as computed by three well-known theories of the distribution of the stresses. The comparison was made in the cases of eleven beams, in the testing of which the necessary observations

[1] Instructor Massachusetts Institute of Technology.

To be presented at a Meeting of The American Society of Mechanical Engineers. All papers are subject to revision.

were taken. Of the eleven, five were tested in the laboratory of Applied Mechanics of the Massachusetts Institute of Technology, and six in the laboratory of the University of Illinois.

4 The reinforcement consisted in each case of one or more longitudinal bars placed near the bottom of the beam, and equal loads were applied at the two points which divided the span into thirds.

5 The three theories employed in making the calculations, all of which assume that at any given section the strain in any fibre is proportional to the distance of the fibre from the neutral axis, will be denoted by *A*, *B* and *C* respectively, and may be described as follows, the notation used in lettering the figures being explained subsequently:

A This theory, which is very extensively employed, makes the assumption that at any given section none of the

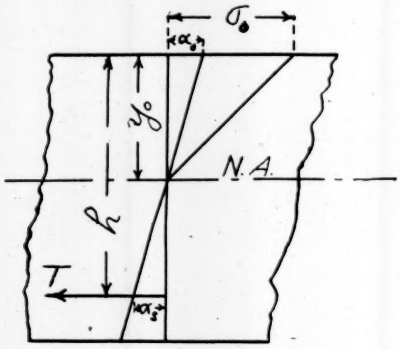

FIG. 1 THEORY A: DISTRIBUTION OF STRAINS AND STRESSES AT A CROSS-SECTION

concrete below the neutral axis can be relied upon to resist tension; and further that the stress is proportional to the strain not only in the steel but also in the concrete.

6 This method is used, by those who employ it to determine (*a*) the position of the neutral axis, (*b*) the stress in the steel, (*c*) the stress in the concrete, and sometimes the shearing stress at the neutral axis; but practically no attempt is made to compute the deflections by it. Nevertheless for the purpose of comparison, deflection formulæ deduced on this basis will be given. The distribution of the strains and stresses at a cross section is shown in Fig. 1.

B This theory, which was proposed by Prof. A. N. Talbot, also makes the assumption that at any given section none of the concrete below the neutral axis can be relied upon to resist tension; but instead of assuming the proportionality of stress to strain in the concrete, the assumption is made that the stress at any fibre can be represented graphically by the corresponding abscissa of a parabola drawn through the neutral axis; the axis of the parabola being at right angles to the section, and its vertex at the end of the abscissa which would represent the crushing strength per square inch of the concrete, were the plot continued to such a height as to correspond to this crushing strength.

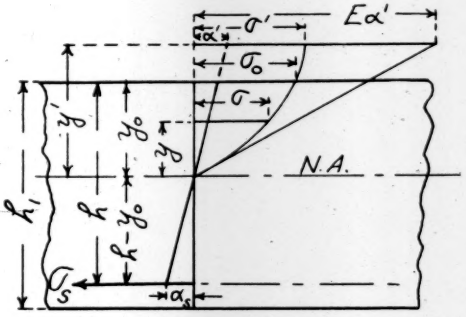

FIG. 2 THEORY B: DISTRIBUTION OF STRAINS AND STRESSES AT A CROSS-SECTION

7 The quantities calculated by this theory are the same as in case *A*, and again deflection formulae will be deduced on this basis for the same reason as there stated. The distribution of the strains and stresses at a cross section is shown in Fig. 2.

C The third theory is that proposed by Mr. Considère. He claims that whereas in a plain concrete beam, the concrete on the tension side cracks when the extension has reached 0.01 to 0.02 per cent, in a reinforced-concrete beam the concrete on the tension side can undergo many times this extension without cracking.

8 Among the tests which he cites in confirmation of this view is the following: He says that he subjected one reinforced-concrete beam to a load that produced in the lower fibre of the concrete an

elongation of 0.063 per cent as determined by measurement, and another such beam to a load that produced in the lower fibre of the concrete an elongation of 0.13 per cent, that he then removed the loads, chipped off the concrete below the reinforcement, and removed the reinforcing bars, after which he smoothed off the lower surface of the remaining portion of the beam and sawed out a concrete plank from the lower side. He says that not only did this plank not fall to pieces, but that on loading it transversely it bore as much as would be expected from a plain concrete plank of the same dimensions.

9 In view of the above, Mr. Considère suggests that the distribution of the stress at a section is as shown in Fig. 3, the compressive strength being represented by the triangle *OABO*, and the tensile strength in the concrete by the trapezoid *OCEFO*, the value of *CD*

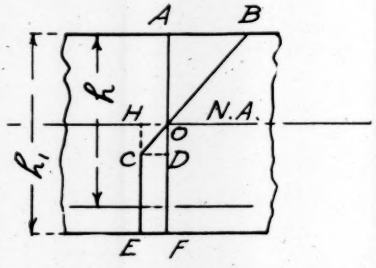

FIG. 3 DISTRIBUTION OF STRAINS AND STRESSES ACCORDING TO CONSIDÈRE'S THEORY

being equal to the yield point of the concrete in tension; and that for greater elongations the tensile strength does not increase.

10 However, inasmuch as the assumption of this distribution would lead to great complexity in the calculations, he proposes as a sufficiently close approximation that for the trapezoid *OCEFO* in Fig. 3 we substitute the rectangle *OHEFO*. In this paper this approximation will be made in obtaining the formulae on the basis of *C*.

11 Before obtaining the formulae needed for making the calculations, the notation used throughout will be explained.

Let α_o = strain in concrete at upper fibre of beam.

α_s = strain in steel reinforcement.

E = ratio of stress to strain in concrete. In B this will denote the initial ratio of stress to strain.

E_s = ratio of stress to strain in steel.

$r = \frac{E_s}{E}$

σ_o = compressive outside fibre stress per square inch in concrete.

σ_s = stress per square inch in steel.

a_s = area of section of steel reinforcement in square inches.

$T = \sigma_s a_s$.

b = breadth of beam.

$n = \frac{ra_s}{b}$

h = distance from top of beam to centre of reinforcement, inches.

h_1 = total depth of beam, inches.

y_o = distance from top of beam to neutral axis.

ρ = radius of curvature of vertical longitudinal section of neutral layer.

W = total load applied.

l = span, inches.

v = deflection at distance x from left-hand support.

v_o = greatest deflection, i. e., deflection at middle.

The above is the notation needed for A .

12 In B the same notation is used with the following in addition:

Let y = distance of any fibre above the neutral axis.

y^1 = distance above neutral axis at which the fibre would be subjected to the crushing strength.

α = strain of fibre at distance y above neutral axis.

α^1 = ultimate compressive strain of concrete.

σ = stress in fibre at distance y above neutral axis.

σ^1 = ultimate compressive strength of concrete.

$q = \frac{\alpha_o}{\alpha^1}$

$n_1 = \frac{3\,r\,a_s}{b\,(3 - q)}$

d_1 = distance above neutral axis to point of application of resultant of compression.

d_2 = distance below top of beam to point of application of resultant of compression.

13 In C the same notation is used with the following in addition:

Let t = yield point of concrete in tension.

$d = h_1 - h$.

FORMULAE

14 Taking up the three theories successively, the formulae needed to make the computation of the values of y_o, σ_s, σ_o and v_o will now be given, the deduction being left to the reader. By Method A:

A

$$y_o = \sqrt{n^2 + 2nh} - n \quad \ldots\ldots (1)$$

$$\sigma_s = \frac{3M}{(3h - y_o)\, a_s} \quad \ldots\ldots (2)$$

$$\sigma_o = \frac{6M}{(3h - y_o)\, by_o} \quad \ldots\ldots (3)$$

$$v_o = -\frac{23\, Wl^3}{1296\, A} \quad \ldots\ldots (4)$$

where

$$A = E\left\{ra_s\,(h - y_o)^2 + \frac{b\,y_o^3}{3}\right\}$$

B

15 In order to find y_o, σ_s, and σ_o by Method B, we have the following equations, all of which include q:

$$y_o = \sqrt{n_1^2 + 2n_1h} - n_1 \quad \ldots\ldots (5)$$

$$\sigma_s = \frac{M}{a_s\left\{h - y_o\frac{4-q}{4\,(3-q)}\right\}} \quad \ldots\ldots (6)$$

$$\sigma_o = \frac{M}{by_o\left\{\frac{3-q}{3\,(2-q)}\,h - \frac{4-q}{12\,(2-q)}\,y_o\right\}} \quad \ldots\ldots (7)$$

Hence before we can find the values of y_o, σ_s and σ_o we need to determine q, and this will have to be done approximately. For this purpose we can use the equation:

$$q^2 - 2q = -\frac{\sigma_o}{\sigma^1} \quad \ldots\ldots (8)$$

16 Plot a curve having $\frac{\sigma_o}{\sigma^1}$'s for abscissae, and q's for ordinates, then using for σ_o a first approximation to its value, determine a first

approximation for q. Then determine a second approximation for δ_o and from it a second approximation for q, etc.

17 In the calculations made here, with the load approximately one-third of the breaking load, the value that has been employed is $q = 0.2$.

18 For the deflection we have

$$v_o = -\frac{23\,Wl^3}{1296\,A} \qquad (9)$$

where

$$A = E\left\{ r\,a_s\,(h - y_o)^2 + b\,y_o^3\,\frac{8 - 3q}{24} \right\}$$

C

19 In order to obtain y_o by Method C we need to solve the equation of the fourth degree in y_o:

$$b\,y_o^4 + 2\left\{ b\,(h + d) + 3\,r\,a_s \right\} y_o^3 - 3\left\{ b\,(h + d)^2 + 6\,r\,a_s\,h - 2\,\frac{M}{t} \right\} y_o^2 + 6\,r\,a_s \left\{ \frac{2\,M}{t\,b} + 3\,h^2 - d^2 \right\} y_o - 6\,r\,a_s\,h \left\{ \frac{2\,M}{t\,b} + h^2 - d^2 \right\} = 0 \qquad (10)$$

20 The solution can be readily effected graphically for any numerical case by writing u equal to the entire left-hand side of the equation, and plotting the resulting curve with y_o's as abscissae and u's as ordinates; then the value of y_o where this curve crosses the axis of abscissae will be the value of y_o desired. Of course the equation has four roots, but the one required can be easily identified as it must give a neutral axis that lies within the section.

21 In solving this equation, some value of t, the tensile yield point of the concrete, must be used. Considère suggests 170 lb. per sq. in. for the concrete used by him, which was about six months old and of a composition of nearly 1 — 2.5 — 2.5.

22 In the calculations made in this paper, $t = 100$ lb. per sq. in. has been used, as the concrete was from 30 to 60 days old and its composition was 1 — 3 — 6. After y_o has been found we can find σ_s and σ_o from the following equations respectively:

$$\sigma_s\,a_s\,\frac{3h - y_o}{3} = M - t\,b\,(h_1 - y)\,\frac{3h_1 + y_o}{6} \qquad (11)$$

$$\sigma_o\,\frac{b\,y_o}{2}\,\frac{3h - y}{3} = M + t\,b\,(h_1 - y_o)\,\frac{2h - h_1 - y_o}{2} \qquad (12)$$

or more easily from the formulae.

$$\sigma_o = \frac{t\,b\,(h_1 - y_o)}{\frac{b\,y_o}{2} - r\,a_s\,\frac{h - y_o}{y_o}} \qquad \text{.....(13)}$$

$$\sigma_s = r\,\sigma_o\,\frac{h - y_o}{y_o} \qquad \text{.....(14)}$$

For the deflection we have

$$v_o = -\frac{23\,W\,l^3}{1296\,A} \qquad \text{.....(15)}$$

where

$$A = E\left\{\frac{r\,a_s\,(h - y_o)\,(2\,h - h_1 - y_o)}{2} + \frac{b\,y_o^{\,2}\,(3\,h_1 + y_o)}{12}\right\}$$

COMPARISON OF THE VALUES OF y_o, σ_s σ_o AND v_o AS COMPUTED BY THE VARIOUS THEORIES WITH THOSE DETERMINED BY EXPERIMENT

23 This comparison is exhibited in the tables. The first five beams were tested in the laboratory of applied mechanics of the Massachusetts Institute of Technology, and for these we have used $E_s = 28{,}000{,}000$ and $E = 2{,}335{,}000$, and hence $r = 12$. The last six beams were tested in the laboratory of the University of Illinois, and for these we have used $E_s = 30{,}000{,}000$, and $E = 2{,}000{,}000$, and hence $r = 15$. All eleven were made of $1 - 3 - 6$ concrete, the ages being given in the tables. All were loaded with two equal loads applied at points dividing the span into thirds.

TABLE I DETAILS OF REINFORCED CONCRETE BEAMS

ALL BEAMS LOADED AT THIRD POINTS

DESIGNATION OF BEAM	AGE DAYS	b INCHES	h INCHES	h_1 INCHES	SPAN FEET	RODS NUMBER	RODS SIZE INCHES ROUND	STEEL a_s AREA IN SQUARE INCHES	p*
Massachusetts Institute of Technology									
A—1	53	8	10	12	11	1	1†	1.00	1.25
A—2	49	8	10	12	11	1	1‡	1.00	1.25
B—3	43	8	10	12	11	2	¾†	1.125	1.41
C—5	35	8	10	12	11	4	½†	1.00	1.25
E—9	54	8	10	12	11	2	⅞‡	1.53	1.91
University of Illinois									
11	65	8	10	11	12	4	½	0.785	0.99
27–'04	63	12	12	13½	14	4	¾†	2.25	1.56
28	60	8	10	11	12	4	¾	1.77	2.22
33	60	8	10	11	12	3	¾	1.325	1.66
35	60	8	10	11	12	3 2	½ ¾	1.473	1.84
45	61	8	10	11	12	3 2	½ ¾	1.473	1.84

* Reinforcement of area above center line of steel, per cent.
† Square.
‡ Twisted.

TABLE 2 DATA FROM TESTS ON REINFORCED CONCRETE BEAMS

ALL BEAMS LOADED AT THIRD POINT

DESIGNATION OF BEAM	BREAKING LOAD*	NEAREST ⅓ LOAD†	INCHES ACTUAL (PLOT) y_0	σ_0	σ_s	DEFLECTION Load Considered W	v_0 (Plot)
Massachusetts Institute of Technology							
A—1	15250	5250	5.4	726	7941	4000	0.0731
A—2	16500	5250	5.3	650	7644	4000	0.0749
B—3	15950	5250	5.5	565	6615	4000	0.0660
C—5	16240	4600	4.6	781	8246	4000	0.1015
E—9	22250	6250	5.1	776	7563	5000	0.0769
University of Illinois							
11	11000	4000	4.8	740	11700	4000	0.175
27–'04	26900	9000	6.8	680	8250	9000	0.162
28	14300	5000	5.8	760	7800	5000	0.141
33	14400	5000	4.9	580	9000	5000	0.137
35	15000	5000	6.0	660	6750	5000	0.100
45	12400	4000	6.0	660	6750	4000	0.150

* Exclusive of weight of beam.
† Used in plots and in calculation for y_0, σ_s, σ_0.

TABLE 3 RESULTS OBTAINED BY EXPERIMENT AND BY COMPUTATION

Massachusetts Institute of Technology Beams: $r = 12$, E = 2,335,000.
University of Illinois Beams: $r = 15$, E = 2,000,000.

Designation of Beam	y_o				σ_o			
	Actual or Plot	A	B $q = 0.20$	C $t = 100$	Actual or Plot	A	B $q = 0.20$	C $t = 100$
Massachusetts Institute of Technology								
A—1	5.4	4.18	4.29	4.94	726	801	760	759
A—2	5.3	4.18	4.29	4.94	650	801	760	759
B—3	5.5	4.36	4.47	5.10	565	774	734	640
C—5	4.6	4.18	4.29	5.06	781	703	666	653
E—9	5.1	4.86	4.97	5.45	776	844	802	810
University of Illinois								
11	4.8	4.15	4.27	4.93	740	670	634	645
27–'04	6.8	5.88	6.01	6.60	680	711	678	690
28	5.8	5.49	5.59	5.98	760	669	638	662
33	4.9	5.00	5.10	5.54	580	722	686	717
35	6.0	5.17	5.28	5.71	660	701	667	675
45	6.0	5.17	5.28	5.83	660	561	534	548

TABLE 4 RESULTS OBTAINED BY EXPERIMENT AND BY COMPUTATION

Massachusetts Institute of Technology Beams: $r = 12$, $E = 2{,}335{,}000$.
University of Illinois Beams: $r = 15$, $E = 2{,}000{,}000$.

Designation of Beam	σ_s				v_o			
	Actual or Plot	A	B $q = 0.20$	C $t = 100$	Actual or Plot	A	B $q = 0.20$	C $t = 100$
Massachusetts Institute of Technology								
A—1	7941	13420	13510	9333	0.0731	0.1163	0.1193	0.0921
A—2	7644	13420	13510	9333	0.0749	0.1163	0.1193	0.0921
B—3	6615	12000	12100	7374	0.0660	0.1076	0.1105	0.0865
C—5	8246	11755	11840	7650	0.1015	0.1163	0.1193	0.0887
E—9	7563	10720	10810	8110	0.0769	0.1105	0.1139	0.0942
University of Illinois								
11	11700	14190	14210	9950	0.175	0.1778	0.1824	0.1500
27–04	8250	11160	11240	8468	0.162	0.1822	0.1878	0.1573
28	7800	8296	8386	6672	0.141	0.1346	0.1400	0.1216
33	9000	10880	10960	8662	0.137	0.1592	0.1647	0.1406
35	6750	9842	9926	7609	0.100	0.1501	0.1549	0.1329
45	6750	7890	7941	5878	0.150	0.1201	0.1240	0.1033

TABLE 5 COMPARISON OF RESULTS

PER CENT OF VARIATION OF RESULTS ON TABLES 3 AND 4 FROM THE ACTUAL VALUES DETERMINED BY EXPERIMENT

DESIGNATION OF BEAM	y_o			σ_s		
	A	B	C $t = 100$	A	B	C $t = 100$
Massachusetts Institute of Technology						
A—1	−22.59	−20.56	−8.52	69.00	70.13	17.54
A—2	−21.12	−19.05	−6.67	75.60	76.74	22.09
B—3	−20.73	−18.73	−7.27	81.41	82.92	11.47
C−5	− 9.13	− 6.74	10.00	42.55	43.53	−7.23
E−9	− 4.71	− 2.55	6.86	41.74	42.93	7.24
University of Illinois						
11	−13.54	−11.04	2.71	21.28	21.45	−14.96
27	−13.53	−11.61	−2.94	35.27	36.24	2.64
28	− 5.35	− 3.62	3.11	6.36	7.51	−14.45
33	2.04	4.09	13.06	20.83	21.78	− 3.76
35	−13.89	−12.00	−4.84	45.81	47.05	12.72
45	−13.84	−12.00	−2.84	16.89	17.65	−12.91
Average...	−12.40	−10.32	0.24	41.52	42.55	1.85

Values less than the actual are called negative.

TABLE 6

PER CENT OF VARIATION OF RESULTS ON TABLES 3 AND 4, FROM THE ACTUAL VALUES DETERMINED BY EXPERIMENT

DESIGNATION OF BEAM	σ_o			v_o		
	A	B	C $t = 100$	A	B	C $t = 100$
Massachusetts Institute of Technology						
A—1	10.33	4.68	4.55	59.09	63.20	25.99
A—2	23.23	16.92	16.77	55.27	59.23	22.96
B—3	36.99	29.91	13.27	63.03	67.44	31.07
C—5	− 9.98	−14.72	−16.39	14.58	17.54	−12.61
E—9	8.76	3.35	4.38	43.70	48.11	22.50
University of Illinois						
11	− 9.46	−14.34	−12.83	1.60	4.23	−14.29
27	4.56	− 0.29	1.47	12.47	15.92	− 2.90
28	−11.98	−16.05	−12.89	− 4.54	− 0.71	−13.76
33	18.28	18.28	23.62	16.20	20.22	2.63
35	6.21	1.06	2.27	50.01	54.90	32.80
45	−15.00	−19.10	−16.97	−19.93	−17.34	−31.15
Average...	6.19	0.88	0.66	26.50	30.25	5.76

Values less than the actual are called negative.

REMARKS AND CONCLUSIONS

24 The results seem to warrant the statement in Par. 1 that "the observations made thus far are not sufficient to furnish the means for determining the actual distribution of the stresses, and hence for the deduction of reliable formulae for the computation of the direct stresses, shearing stresses, diagonal stresses, deflections, position of the neutral axis, etc., under a given load." It follows therefore that whichever of the theories is adopted for practical use, it can be regarded only as a sort of working hypothesis.

25 It seemed therefore desirable to compare the results of these three well-known theories with those obtained by experiment. This comparison can best be made by a detailed study of the tables, especially Table 5 and Table 6.

26 However, it seems plain, as far as the evidence of these eleven tests goes, that in deducing the values of y_o and σ_s theory C gives results much nearer those determined by experiment than either A or B, and the same is true to a lesser degree in the case of v_o, whereas the differences are not so marked in the case of σ_o.

27 It also seems hopeless to obtain a reliable deflection formula without taking into account the tension in the concrete.

28 Of course the computations are more easily made when A is used rather than B or C, but in the cases of B and C the complexity is not so great when designing a beam as when determining the stresses in a given beam under a given load.

PUMP VALVES AND VALVE AREAS

By A. F. Nagle, South Bethlehem, Pa.

Member of the Society

There has grown up a custom of specifying in water works pumping engine specifications that the area through the valves shall exceed the area of the plunger by a certain percentage thereof, varying from 25 to 125 per cent of increase. It is my purpose to demonstrate that the probable intent of the writer of this clause in the specification is not obtained in practice, namely, a low velocity through the valve and consequent low loss of head.

2 The above form of expression, namely, proportioning the valve area to the plunger area, is defective because (*a*) it fails to distinguish between the valve-seat area and the circumferential area of the valve at an assumed or specified lift; (*b*) it leads to an absurdity unless coupled with the length of stroke and the number thereof.

3 To the first criticism it may be replied that the engine builder interprets the clause to mean the net area through the valve seats, but the city's engineer occasionally requires the circumferential area. To this the builder will not seriously object, for he simply increases the *possible* lift, knowing very well that "it will never go there."

4 The second criticism can be best illustrated by the following example: Compare two pumps, each making 25 r.p.m. one having a plunger 6½ in. in diameter by 60 in. stroke, and the other a plunger of 13 in. diameter by 15 in. stroke. Precisely the same volume of water passes through the two pumps, yet the rule laid down in the specification would require for one pump four times the valve area of the other.

5 What is the real purpose in specifying anything at all about valve area? Evidently the same that is sought in limiting the plunger travel per minute, and founded upon the law that in a pumping engine, low velocities of water are conducive to low cost of operation, but proportionately great cost of construction, and conversely,

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high velocities imply high cost of operation but lower cost of construction; hence, very properly, the buyer should specify the maximum velocity he will accept.

6 Briefly, it may be said that city water works engines are now quite generally limited to about 250 ft. of plunger travel per min., although frequent attempts are made with special designs to increase this travel to nearly twice this amount. This plunger travel with 5 to 1 connecting rod, entails a maximum plunger velocity of 6.67 ft. per sec., and the *head due to this velocity* is 0.30 lb. per sq. in. It is desirable to speak of the maximum plunger velocity rather than the mean, because that governs the maximum valve area to be provided.

7 The fluid losses within a pump may be divided as follows:

a Velocity head due to plunger velocity, varying from zero to the maximum above cited. This loss may be ignored however, since with well rounded plunger ends and rounded water passages, the accelerating head of the fluid column during the first half of the stroke is conserved by its retarding force during the second half.

b Friction head due to surface contact. As the main parts of a pump are comparatively large, the velocities are low: and the lengths of contact being short, this friction-head is equal to a velocity-head for only about 50 diameter lengths, and becomes so small as to be negligible.

c Velocity head through the valves. This, whatever its amount, is a total loss because the energy of the issuing streams is destroyed in eddies as it enters the large valve or pump chambers. To keep this head low is the purpose of the specification stating that the valve area shall exceed the plunger area by a certain amount.

8 Let us assume that valve area means valve-seat area, and pass on to the study of the valve. A pump valve consists essentially of three elements (*a*) a fixed seat, (*b*) a movable valve, (*c*) a spring. The most important of these is the spring, and yet on this point all specifications are silent. Is this because the writer of the specification knows nothing about the subject? In a general way, it is obvious that a spring may be so stiff that on the suction stroke, where only atmospheric pressure is available, the valve will not open at all; or it may be so light that it will nearly float in its place and will close only with the return stroke of the plunger. Between these two extremes, is there not an ascertainable strength of spring which will

allow the valve to close promptly without shock and yet require for lifting force but a small percentage of the total pressure in the pump? So far as I am aware, this problem has not been stated and solved in any publication, but is left for each pump builder and user.

9 The pressure of the spring per square inch of the inside seat area seems to me to be the force that causes the rate of flow of the water through the valve. In my experiments of 1875 (see Vol. 10 of Transactions) with a Cornish double-beat valve, this hypothesis did not hold, that is, the velocity through the valve was from 60 to 90 per cent greater than that due to the pressure of the valve: in other words, the valve did not rise as high as theory would demand,

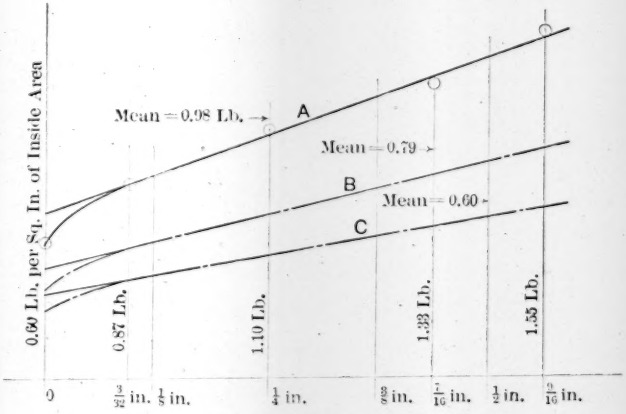

FIG. 1 DIAGRAM SHOWING VARIATIONS OF TENSION

LINE A IS THE ACTUAL TENSION OF A SPRING AT VARIOUS POINTS OF LIFT. LINES B AND C SHOW ESTIMATED TENSIONS AT DIFFERENT LIFTS WITH AN INITIAL TENSION OF 0.4 LB. AND 0.3 LB.

but I think the deviation may be attributed to the large curvature given the upper passage. In extensive experiments recently made by the Bethlehem Steel Company with a large flat-hinge, or flap, valve to be used in the Baltimore sewage pumps, the hypothesis held very well at the beginning of the lift and fell off only about 10 per cent at full lift. These experiments also confirmed the Providence experiments in that the varying lift of the valve follows closely the varying velocities of the plunger, except as it is modified by increased weight or spring tension.

10 I shall therefore assume that in a flat rubber pump valve held down by a spring (*a*) the velocity of the water is that due to pressure

per square inch of the inside valve area, (*b*) the area of discharge is the net circumference of the inside of the seat multiplied by the lift.

11 The well-known formula for the velocity of flow in feet per second is

$$V = 8.025 \sqrt{h} \qquad [1]$$

(*a*) where h is the head of water in feet. As 2.31 ft. of water 1 sq. in. in area weighs 1 lb., the formula can be changed to

$$V = 12.23 \sqrt{p} \qquad [2]$$

(*b*) where p is the pressure of spring per square inch of inside area

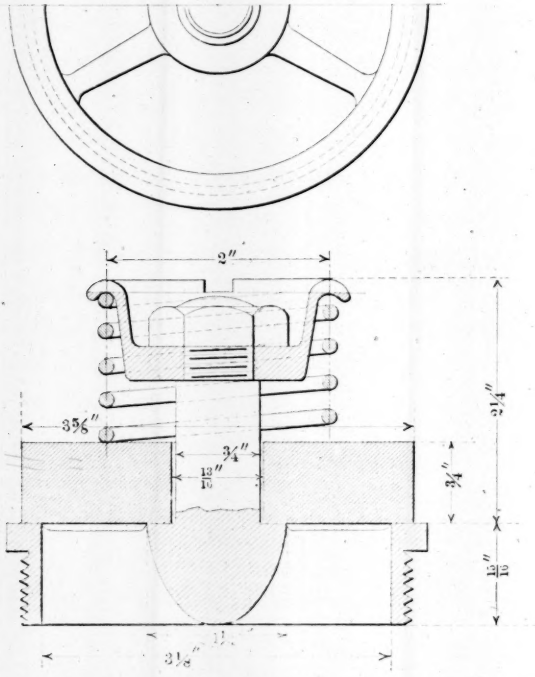

FIG. 2 SECTION OF A STANDARD MAKE OF PUMP VALVE

FREE LENGTH OF SPRING IS $1\frac{9}{16}$ IN., NO. 12 B. W. G. SPRING BRASS

of valve. Tables 1 and 2, computed from Formula 2, may be convenient in studying this subject.

12 *Springs.* The springs in common use vary from 0.40 to 0.60 lb. per sq. in. of inside valve area at the beginning of the lift, and as they are comparatively short (about $1\frac{5}{8}$ in. closed), they tighten up

TABLE 1 CONVERSION OF VELOCITY INTO PRESSURE

Velocity Ft. per Sec.	Pressure Pounds per Sq. In.	Velocity Ft. per Sec.	Pressure Pounds per Sq. In.	Velocity Ft. per Sec.	Pressure Pounds per Sq. In.
4	0.107	8	0.430	12	0.967
5	0.168	9	0.544	13	1.135
6	0.242	10	0.672	14	1.317
7	0.329	11	0.813	15	1.512

TABLE 2 CONVERSION OF PRESSURE INTO VELOCITY

Pounds Pressure	Velocity Ft. per Sec.	Pounds Pressure	Velocity Ft. per Sec.	Pounds Pressure	Velocity Ft. per Sec.	Pounds Pressure	Velocity Ft. per Sec.
0.15	4.74	0.50	8.65	0.85	11.27	1.40	14.47
0.20	5.47	0.55	9.07	0.90	11.60	1.50	14.98
0.25	6.11	0.60	9.47	0.95	11.92	1.60	15.47
0.30	6.70	0.65	9.86	1.00	12.23	1.70	15.94
0.35	7.23	0.70	10.23	1.10	12.82	1.80	16.40
0.40	7.73	0.75	10.59	1.20	13.39	1.90	16.85
0.45	8.20	0.80	10.94	1.30	13.94	2.00	17.29

TABLE 3 RATIO OF PRESSURES, VELOCITIES, AND LIFT OF VALVE, LINE A

Lift Inches	Tension Pounds per Sq. In.	Velocity Ft. per Sec.
At Start	0.60	9.47
$\frac{3}{32}$	0.87	11.40
$\frac{1}{4}$	1.10	12.82
$\frac{7}{16}$	1.33	14.10
$\frac{9}{16}$	1.55	15.23

TABLE 4 RATIO OF PRESSURES, VELOCITIES AND LIFT OF VALVE, LINE B

Lift Inches	Tension Pounds per Sq. In.	Velocity Ft. per Sec.
At Start	0.40	7.74
$\frac{3}{32}$	0.58	9.31
$\frac{1}{4}$	0.73	10.47
$\frac{7}{16}$	0.88	11.51
$\frac{9}{16}$	1.03	12.41

TABLE 5 RATIO OF PRESSURES, VELOCITIES AND LIFT OF VALVE, LINE C

Lift Inches	Tension Pounds per Sq. In.	Velocity Ft. per Sec.
At Start	0.30	6.70
$\frac{3}{32}$	0.44	8.07
$\frac{1}{4}$	0.55	9.07
$\frac{7}{16}$	0.66	9.97
$\frac{9}{16}$	0.77	10.83

rapidly as the valve rises. Fig.1, Line A, illustrates this rate of increase taken from a new spring. The apparently needlessly stiff springs are used (*a*) to provide against the relaxation sure to occur with all bronze springs; (*b*) to allow for the lengthening of the spring as the rubber valve wears away.

13 Fig. 2 shows a standard pump valve used by a prominent builder. These valves run from $2\frac{1}{2}$ in. to $3\frac{3}{4}$ in. inside diameter. Table 3 is made up from Fig. 1, with the velocities computed by Formula 2. A larger lift than $\frac{9}{16}$ in. is not generally allowed for, as the valve is not expected to rise higher or even as high as this, and considering the increased tension of the spring one would not expect it.

14 *Other spring tensions.* In Fig. 1, Line A represents the actual tension of a spring at various points of lift. If the same type of spring were made of smaller wire, its varying tensions at different lifts would be proportional to the initial tension. Lines B and C show these estimated tensions at different lifts with an initial tension of 0.40 lb. and 0.30 lb., respectively; Tables 4 and 5 give the tension at these lifts, together with the velocities corresponding thereto.

15 *Valve Lift.* If it is true that the spring tension governs the velocity of water through the valve, we can readily find the lift of a valve under specified conditions. Knowing the volume of water passing through the valve per second, the inside net circumference multiplied by its lift and the velocity must equal the volume. By formula, $P \times V_m = C \times L \times v_m \times N$, where

P = plunger area in square inches.
V_m = maximum velocity of plunger in feet per second; which is 1.60 × the mean velocity.
C = net circumference of valve seat, inches.
L = lift of valve, inches.
v_m = maximum velocity of water at Lift L, found by aid of the diagram, Fig. 1.
N = number of valves.

or

$$L = \frac{P \times V_m}{C\ N\ v_m}$$

Contractions at sharp corners and angular turns make this calculation inexact, but the method will be found correct enough for the comparisons in this paper, and is the only practical method in the present state of knowledge on this subject.

16 *Comparisons of valve areas and springs.* To illustrate my views, let us take the case of a vertical triple-expansion crank and flywheel pumping engine, having each plunger 34 in. in diameter by 60 in. stroke, making 25 r.p.m.—practically a 25,000,000-gal. engine. Plunger travel = 4.167 ft. per sec., and maximum plunger velocity = 6.67 ft. per sec. Assume the pump valve to be $3\frac{3}{4}$ in. inside diameter with 5 ribs, leaving a net area of 8 sq. in., and a net circumference of 10.53 in. The theoretical lift of this valve, to give the same area on the circumference as through the ribs, would be 0.76 in. Let us assume a valve-seat area equal to 150 per cent of the plunger area and an initial spring tension of 0.60 lb., and ascertain the number of valves, their lift, velocities at various points and loss of efficiencies. Then let us make the same calculations for a valve-seat area equal to the plunger area, with an initial spring tension of (*a*) 0.40 lb. per sq. in.; (*b*) 0.30 lb. per sq. in. The results are given in Table 6.

TABLE 6 LOSS OF EFFICIENCIES, ETC.

Initial Spring Pressure Pounds	Valve Seat Area Per Cent	Number of Valves	Lift of Valves Inches	Velocities in Feet per Second			Loss of Efficiency Per Cent
				Plunger	Valve Seat	Valve	
1	2	3	4	5	6	7	8
0.60	150	170	$\frac{1}{4}$	0–6.67	0–4.44	9.47–12.82	2.45
0.40	100	114	$\frac{7}{16}$	0–6.67	0–6.57	7.74–11.51	1.97
0.30	100	114	$\frac{1}{2}$	0–6.67	0–6.57	6.70–10.37	1.50

17 *Explanations.* Column 4. The plunger area (908 sq. in.) multiplied by its maximum velocity (6.67 ft. per sec.) must equal number of valves (170 or 114) multiplied by the net circumference (10.53 in.), its lift L, and the maximum velocity at its highest lift. This is a trial process, but easily found after one or two trials. Taking the first case, we would have

$$908 \times 6.67 = 170 \times 10.53 \times 12.82, \text{ or } L = 0.264 \text{ in.}$$

Second case:

$$908 \times 6.67 = 114 \times 10.53 \times L \times 11.51, \text{ or } L = 0.438 \text{ in.}$$

Third case:

$$908 \times 6.67 = 114 \times 10.53 \times L \times 10.37, \text{ or } L = 0.50 \text{ in.}$$

18 Column 5. While the crank velocity may ordinarily be taken as the maximum plunger velocity with a connecting rod five times the crank, its maximum velocity is 1.019 times that of the crank or, maximum velocity $= 5 \times 3.1416 \times 25 \times 1.019 \div 60 =$ 6.67 ft. per sec. We can also take the mean travel of the plunger and multiply it by 1.60 to find the maximum velocity.

19 Column 6 is self-evident.

20 Column 7. These velocities are obtained from Tables 3 to 5 and Fig. 1, and were computed in the manner already described.

21 Column 8. To get an expression for the effect of strong *vs.* light springs upon the economical working of a pump, I have assumed a pump working under a total head of 80 lb. per sq. in. and computed the *mean* pressure required to operate the valve, calling this ratio of pressures its loss of efficiency.

22 A careful examination of the diagram of spring compression, Fig. 1, shows that at the beginning of the lift the spring did not assume its full uniform resistance. It took nearly $\frac{3}{32}$ in. of motion to tighten it uniformly. I think this is due to the fact that these single coil-wound springs are always a little stiffer on one side than the other, thus canting the valve to an oblique position to conform itself to the center line of resistance of the spring. I have taken as mean pressures of the springs slightly more than the mean of the two extreme positions, because the times during which the different pressures prevail are not equal. Mean pressures were taken as follows:

Case A, maximum tension = 1.10 lb. at $\frac{1}{4}$-in. lift, mean pressure = 0.98 lb..

Case B, maximum tension = 0.94 lb. at $\frac{7}{16}$-in. lift, mean pressure = 0.79 lb.

Case C, maximum tension = 0.72 lb. at $\frac{1}{2}$-in. lift, mean pressure equals, 0.60 lb.

As this mean pressure exists during both strokes, it must be multiplied by two to find its ratio to the effective head of 80 lb., operating only during one stroke. Thus the values given in Column 8 are found.

23 *Discussion and Recommendation.* A study of the figures given in Table 6 shows that the proper place to look for "loss of head" in a pump is in the spring tension, and not in the valve-seat area. As long as the maximum velocity through the seat does not exceed that through the valve, it does not add to the total loss of head. The only

reason for having a large number of pump valves and a large inside diameter is to keep the lift down, basing judgment on the number of reversals per minute.

24 We have seen that a 3¾-in. valve, having a net area of 8 sq. in., needs 0.76 in. lift to give the same area at the circumference as through the seat. If there were no rib obstruction, 25 per cent of the diameter of a circle gives the height to which a valve must lift to give a circumferential opening equal to its area. Because of the ribs, we need but 20 per cent of the diameter for the lift; and with the lightest spring *C*, a maximum lift of ½ in. or 13 per cent of the diameter was sufficient to discharge the required volume of water. It will be good construction to limit the lift of a pump valve of this type to, say, 15 per cent of its internal seat diameter. The spring will not allow it to rise to that height, but it is a safe limit for a stop.

25 The place to begin the study of proportions of a pump is at the *spring* of the valve. *Make* a sample spring of such diameter and length and strength as you may think desirable, and by *experiment* construct a diagram of its rate of compression, as in Fig. 1. Now you can find the maximum velocity at an assumed lift and proceed in the manner already pointed out. The spring would be improved, that is, it would not tighten up so rapidly when compressed, if it could be made somewhat longer than present practice, but this is not practicable, as it would enlarge the valve chamber, where the valve-cage design is used.

26 The largest and weakest castings within a pumping engine are the valve chambers and anything that can be done to reduce them to the minimum size permissible is good engineering. I think the line of study I have pursued will indicate that the total valve-seat area in this type of engine need not be more than the plunger area. That rule, if adopted, would reduce the diameter of the valve chambers to an appreciable extent, probably 10 per cent, and this is well worth saving.

27 The number of valves saved by the construction recommended (about 33 per cent) is also worth saving. No loss whatever would be entailed and a part of the money saved could be expended in making a better spring. I would make the spring of steel, if possible, oil-tempered, and protected against corrosion by copper electroplating. Then I would have all springs tested and brought to a like tension under a rigid specification. With these improvements I believe that a little better pumping engine than we now have could be obtained at a little smaller cost.

A REPORT ON CAST-IRON TEST BARS

By A. F. Nagle, Bethlehem, Pa.
Member of the Society

In machinery castings as well as in cast pipes, separate bars are cast and subjected to tensile or transverse stress to the breaking point, these results being used as evidence of compliance with the contract specifications. The writer has examined a large number of such test bars for castings used in the Baltimore sewage pumps and here reports the results of this examination and study. Perhaps the most important conclusion is that the test bar is not to be regarded with too much confidence as indicative of the exact strength of the casting.

2 All transverse bars were nominally 2 in. by 1 in. by 24 in. centers. They were cast from two patterns in one mold, made in the same kind of sand as the main casting. The flask was inclined about 30 deg. There was but one gate for the two bars, with suitable risers. The iron for the bars was poured from a small ladle of iron taken as near as might be from the middle of the pour of the main casting. The breaking loads were corrected for varying dimensions of the bars by the formula $W' = \frac{Wbd^2}{2}$, where b and d are the actual dimension, W the actual breaking load, and W' the corrected load or weight. These results are used throughout this paper. The deflections were not corrected.

3 The tensile bars, $1\frac{5}{8}$ in. by 6 in., were cast upright in the same mold as the main casting, within three or four inches thereof, and connected by an upper and lower gate. The tensile bars were turned to $1\frac{1}{2}$ in. diameter and threaded, and the middle portion reduced to 1.129 in. diameter, which is equal to 1 sq. in. area. Table 1 gives the results of the chemical analysis of the several bars tested.

4 From August 5, 1907 to April 4, 1908 there were made 67 *single* tensile bars and the same number of pairs of transverse bars, and the

To be presented at a meeting of The American Society of Mechanical Engineers. All papers are subject to revision.

TABLE 1 ANALYSIS OF CAST-IRON TEST BARS

BARS USED IN I. P. BED PLATE CAST NOVEMBER 21, 1907, AND I. P. FRAME, CAST NOVEMBER 26, 1907 FOR BALTIMORE SEWAGE PUMPS

DATE CAST	TOTAL CARBON	GRAPHITE CARBON	COMBINED CARBON	MANGANESE	PHOSPHORUS	SULPHUR	SILICON	TENSILE STRAIN	TRANSVERSE LOAD	DEFLECTION
Nov. 21, 1907.....	3.580	2.830	0.75	0.79	0.485	0.081	1.59	24,900	2440	0.49
Nov. 26, 1907.....	3.396	2.736	0.66	0.38	0.459	0.124	1.91	22,000	2075	0.40

average of the latter was used in this record. From April 4 to December 19, 1908 there were made 91 *pairs* of tensile bars and an equal number of pairs of transverse bars, and each piece of the pair is recorded, instead of the average.

5 Of these 249 tensile bars and their corresponding transverse bars, 32 sets—26 flat and 6 round—were rejected for defects due to blow-holes and four tensile bars were too hard to bear threading, but the companion piece was used in this record.

6 Of the 217 specimens here recorded, 42 are designated as abnormal, that is, the ratio between the tensile and the transverse bars was either considerably greater or smaller than the average.

7 By referring to Table 2, it will be seen that of the 175 specimens of cast iron running from 20,000 to 30,000 lb. tensile strength, the ratio of tensile to breaking loads is practically 10 to 1 and the deflection 0.45 in.

TABLE 2 COMPARISON OF CAST-IRON TEST BARS

NUMBER OF SPECIMENS	LIMIT OF BREAKING LOAD OF TRANSVERSE BARS	BREAKING LOADS POUNDS		DEFLECTION INCHES	RATIO OF TENSILE TO TRANSVERSE
		Transverse	Tensile		
29	2000 to 2200	2065	21,630	0.43	10.47 to 1
36	2200 to 2400	2289	22,940	0.45	10.02 to 1
51	2400 to 2600	2523	24,880	0.47	9.86 to 1
43	2600 to 2800	2756	26,500	0.49	9.61 to 1
16	2800 to 3000	2894	28,460	0.49	9.83 to 1
175	Averages	2383	23,732	0.45	9.96 to 1

NOTE.—Transverse Bars, rough 2 in. by 1 in. by 24 in. centers; tensile bars, turned 1.129 in. diameter (1 sq. in. area).

8 Table 3 gives 25 abnormal cases where this average ratio is as high as 12.56 to 1 with a deflection of 0.43 in.; also 17 abnormal cases where this average ratio is as low as 7.91 to 1 with a deflection of 0.44 in. And yet the average of both normal and abnormal bars was again very nearly 10 to 1—10.07 to 1.

9 Breaking loads, presumably alike, varied in pairs of transverse bars, and also in pairs of tensile bars, as follows:

Out of 65 pairs of flat or transverse bars,
14, or 22 per cent, average variation 18 per cent.

TABLE 3 COMPARISON OF CAST-IRON TEST BARS

ABNORMAL RESULTS

NUMBER OF SPECIMENS	LIMIT OF BREAKING LOAD OF TRANSVERSE BARS	BREAKING LOADS POUNDS		DEFLECTION INCHES	RATO OF TENSILE TO TRANSVERSE
		Transverse	Tensile		
	Above 10 to 1 ratio				
10	2000 to 2200	2088	27,143	0.41	12.95 to 1
10	2200 to 2400	2294	28,530	0.43	12.44 to 1
4	2400 to 2600	2436	29,600	0.49	12.15 to 1
0	2600 to 2800	00	00		
1	2800 to 3000	2890	34,000	0.45	11.76 to 1
25	Averages	2258	28,365	0.43	12.56 to 1
	Below 10 to 1 ratio				
1	2000 to 2200	2105	17,600	0.50	8.36 to 1
4	2200 to 2400	2359	18,825	0.41	7.98 to 1
7	2400 to 2600	2487	18,814	0.43	7.57 to 1
3	2600 to 2800	2656	21,230	0.45	8.00 to 1
2	2800 to 3000	2969	24,500	0.47	8.25 to 1
17	Averages	2521	19,954	0.44	7.91 to 1

17, or 26 per cent average variation 5.4 per cent.
34, or 52 per cent average variation less than 2 per cent.
Out of 65 pairs of round or tensile bars,
22, or 34 per cent average variation 15 per cent.
20, or 31 per cent average variation 5.5 per cent.
23, or 35 per cent average variation less than 2 per cent.

61 other pairs of flat bars, which had only one companion tensile bar, varied in about the same ratios.

10 Two special flat bars and two special round bars, cast in one mold, one gate and at one pour, varied as follows:

2 flat bars, 12 per cent.
2 round bars, 7 per cent.

11 In order to get some more definite information on these variations, if possible, I had a pair of transverse and a pair of tensile bars made and cast in the same mold, and while the average ratio of tensile to transverse strength was again nearly 10 to 1, as shown in Table 4, the same type of bars again varied 12 per cent and 7 per cent respectively as shown in Par. 10.

12 I have no satisfactory explanation for the great variations in these test bars, and we can only accept the fact that mathematical uniformity in strength of cast-iron bars is not found in the present

TABLE 4 COMPARISON OF CAST-IRON TEST BARS

SPECIAL, TWO SETS CAST IN SAME MOLD AT SAME TIME

NUMBER OF SPECIMENS	LIMIT OF BREAKING LOAD OF TRANSVERSE BARS	BREAKING LOADS POUNDS		DEFLECTION INCHES	RATIO OF TENSILE TO TRANSVERSE
		Transverse	Tensile		
1		2350	23,000	0.50	9.79 to 1
1		2100	21,470	0.45	10.21 to 1
2	Average	2225	22,235	0.47	10.04 to 1
217	All Averages	2380	23,970	0.45	10.07 to 1

state of the art. To anyone questioning the results I can only say from my own knowledge of the circumstances that the personal equation did not enter into them.

13 Careful observation of broken bars did not show that the so-called "skin of the metal" was of any appreciable thickness, and the metal was remarkably homogeneous throughout. The tensile bars being turned, the skin, if there were any, of course disappeared. It is my opinion that the skin adds practically nothing to the strength in either transverse or tensile bars, other causes, though obscure, producing far greater deviations.

14 Although many castings were condemned for physical defects, such as blow-holes, shrink-holes, sand-washes, and shifting of cores, not a single case of cold-shut was discovered. This is in marked contrast with the writer's experience on similar work in other foun-

dries. Excepting a number of steam valves, which were of iron too soft for their purpose, but one large casting, a discharge air chamber weighing 16,000 lb., was condemned for being of unsatisfactory iron. In this case the iron was coarse-grained and brittle, and was required to stand at least 23,000 to 24,000 lb. To remove all doubt that the test bars were truly representative of the iron in the main casting, two tensile bars were cut out of a large flange, which had been at the bottom of the mold. These, from the most favored part of the casting, as will be seen, stood but about 17,350 lb.; 90 per cent of that revealed by the test bars. In this case there was a remarkable agreement among these pairs of bars.

15 It may be interesting to apply these results to the formula for the strength of cast-iron beams subjected to similar stress. The commonly used formula (Kent, page 268) is

$$R = \frac{3\,Pl}{2\,bd^2}$$

TABLE 5 TEST BARS FROM CONDEMNED CASTING

Breaking Loads Pounds		Deflection Inches
Transverse	Tensile	
1,968	19,800	0.35
2,019	19,000	0.50
	Cut out of flange	
	17,000	
	17,700	

where R is called the modulus of rupture, or stress per square inch of extreme fiber.

P = load at center.
l = length in inches between supports.
b and d = breadth and depth, respectively in inches.

Making the proper substitutions, we have $R = \frac{3 \times 2380 \times 24}{2 \times 2 \times 1 \times 1}$, or $R = 42{,}840$ lb. This is not the correct figure, however, for the extreme fiber stress: we know this cannot exceed the tensile strength, which we have found to be 23,732 lb.

16 I think it is better to use D. K. Clark's formula, given on page 507 of his Engineers' Tables, etc., $S = \frac{Wl}{1.155\,bd^2}$, where S = extreme

fiber stress, or tensile strength. If we use the tensile strength found in these tests as 23,732 lb., the breaking load W would become $\frac{23{,}732 \times 1.155 \times 2 \times 1}{24} = 2284$ lb., the actual breaking load being 2383 lb. As this is within 4.3 per cent of the average found in these tests, this formula, using the tensile strength for the extreme fiber stress, seems to me to be the more intelligible and dispenses with the "coefficient of rupture."

17 "Mr. Barlow found by experiment that for 1-in. square bars of cast iron, the breaking weight in tons [2240 lb., I presume] was expressed by the formula $W = \frac{bd^2}{l} \times 13.6$, and Mr. Robert Stephenson arrived by experiment at exactly the same coefficient." Clark, page 561.

TABLE 6 CIRCULAR TEST BARS CAST IN VERTICAL DRY-SAND MOLD

Bar Mark	Breaking Loads Pounds		Deflection Inches	Value of W By Formula Pounds	Original Diameter Inches
	Transverse	Tensile			
H	3344	23,070	0.15	2948	1.305
H	3344	23,754	0.15	3036	1.305
X	3026	24,670	0.12	3153	1.300
1	2	3	4	5	6

18 If we should substitute the value for W found in these tests we would have W, or $\frac{2383}{2240}$, $= 1.064$ tons $= \frac{2 \times 1}{24} \times$ a constant, or, constant $= 12.77$, which is within 7 per cent of the coefficients found by Barlow and Stephenson.

19 Since the foregoing was written I have had the opportunity to observe two circular test bars nominally 1¼ in. in diameter by 15 in. long, with 12-in. centers. These bars were cast from two patterns in one vertical dry-sand mold and poured from a small ladle, first one and then the other, with the results shown in Table 6.

20 The tensile bars were taken from the bottom end of the broken test bar, but I do not know whether H or X was poured first. The first tensile bar H had a small air hole, which being corrected for added 7 per cent to its tensile strength, and this is also given in the table. A second bar was then turned up from the immediate joining piece with the result recorded first in the table. The turned bars

were 0.937 in. in diameter. Column 6 gives the original diameter. Column 2 was found by reducing the actual breaking loads in the ratio of the cubes of the diameters, and Column 3 was reduced to the square inch area. Why the transverse breaking loads should vary 10 per cent and the tensile bars 4 to 7 per cent the opposite way, a total variation of 14 to 17 per cent, I leave to the reflection of the reader. If we apply Clark's formula for the breaking weights for circular bars, $W = \frac{0.7854 \times d^3 \times S}{l}$ we find the values given in Column 5.

21 In this age of economic production, the cost of these turned tensile bars is frequently objected to by the manufacturer. While blowholes seem to be more frequent in flat transverse bars than in round attached tensile bars, the latter seem liable to a greater abnormal hardness, for which I have no explanation. Some indication of the toughness of cast iron can be seen in its deflection, which is not revealed in a direct tensile pull. I should, therefore, be satisfied with two or three transverse test bars 2 in. by 1 in. by 24 in. centers, and a deflection record poured, as near as may be, from the middle of the pour of the main casting, as giving a fair indication of the iron in the main casting, but mathematical exactness cannot be looked for as yet.

22 If we wish to know approximately the corresponding tensile strength of the iron, we can multiply the breaking load of the 2 in. by 1 in. by 24 in. flat bar by 10. If the test bar is of 1¼-in. diameter by 12-in. centers, its breaking load should be multiplied by 8, to obtain the approximate tensile strength. The general rule seems to be, that where both flat bars agree in breaking loads, the tensile strength is 10 to 1 of the breaking load, but where they differ, the 10 to 1 ratio does not hold. A better practice, therefore, might be to cast three round transverse bars and accept the two that agree, if each is round, as a fair sample of the iron, dispensing with the tensile bars. This concession to the manufacturer, I believe, would entail not only no loss to the city's interests, but a positive gain.

TESTS ON A VENTURI METER FOR BOILER FEED

By Prof. C. M. Allen, Worcester, Mass.
Member of the Society

A reliable and accurate hot-water meter has been in demand for a good many years. The principle of most of the cold-water meters, where there are moving parts in the water, is not at all adaptable for hot-water work. A hot-water meter for boiler-feed purposes must stand not only the variation in temperature but also considerable variation in pressure, and quite often it has to stand a certain amount of water-hammer, this depending somewhat upon the style and condition of pump used. The Venturi meter, having no moving parts to get out of order and being of material which will stand the ordinary corrosive effects, should make a reliable hot-water meter.

2 The object of these tests was to determine the accuracy of a Venturi meter to be used for measuring boiler feed under a great variety of conditions. The plan was to make a complete series of tests upon a small Venturi meter under all the probable conditions that would ever be met in boiler room practice. The tests were made under varying temperatures and velocities; under varying pressures, intermittent and steady; using a triplex power pump in good condition, and with one plunger out of commission, a duplex steam pump in good condition and in poor condition, and an injector.

3 The meter was installed in the steam engineering laboratory of the Worcester Polytechnic Institute and set up the way most convenient not only for weighing the water passing through the meter, but also for heating the water before it went through the meter, and pumping it in in various ways. The meter used was built by the Builders' Iron Foundry of Providence, R. I., and is what is ordinarily called a 2-in. meter, the upstream and downstream ends being 2 in. in diameter and the throat $\frac{3}{4}$ in. in diameter. The main part of the meter is of cast iron and the internal portions are lined with brass. Surrounding the upstream end and throat are annular chambers between

To be presented at a meeting of The American Society of Mechanical Engineers. All papers subject to revision.

the brass sleeve and the iron casing. Six holes are drilled through the brass lining into these annular chambers at about equal distances around the circumference, in order to give the actual pressure heads in the meter at both throat and upstream end. From the outside of these annular chambers were pipe connections to a manometer tube which consisted of a glass U-tube containing mercury. There were the necessary valves and pet-cocks to manipulate the meter, blowing out the air whenever it accumulated. The general layout of the apparatus is shown in Fig. 1.

4 The apparatus was set up so that the meter could be supplied from a 1¼-in. metropolitan injector, a 4½ in. by 2¾ in. by 4 in. duplex pump or 4 in. by 5¾ in. triplex power pump; or from a pressure tank supplied from the city mains or by a large duplex pump. These pumps were arranged to take their suction from a pit 12 ft. long, 6 ft. wide and 4 ft. deep, directly beneath the Venturi meter. A 1-in. steam line was put in to heat the water. There being about 300 cu. ft. in the supply pit, a very even temperature could be maintained. In order to keep the discharge from the pumps constant, a suction well was supplied, kept at constant level by an additional pump from the main pit. The discharge from all the pumps used was carried up a vertical 2-in. pipe, at the top of which was an air chamber 4 in. in diameter and 3 ft. long, with a valve so inserted that it might be cut out whenever desired. From this vertical pipe ran a line containing the Venturi meter, a thermometer-well for determining temperatures, and a valve for throttling water in order to get any desired pressure. At the end of this line was a swinging end that discharged into either of two 5000-lb. weighing tanks.

5 The first tests were made with cold water in order to determine the coefficient. These were made with steady pressure, securing a very constant flow. Water was run through the meter until the conditions had become constant. One tank was weighed while the water was being discharged into the other. The tests were started by diverting the discharge into the weighing tank and taking the time. Readings of the Venturi meter were taken every thirty seconds for low velocities and every minute for higher. The tests were ended by diverting the discharge into the other tank, taking the time and weighing.

6 When hot water tests were made, it was found that a certain amount of water evaporated; evaporation tests were therefore made, which proved this amount to be a negligible quantity.

7 In order to compare the workings of the meter under the various

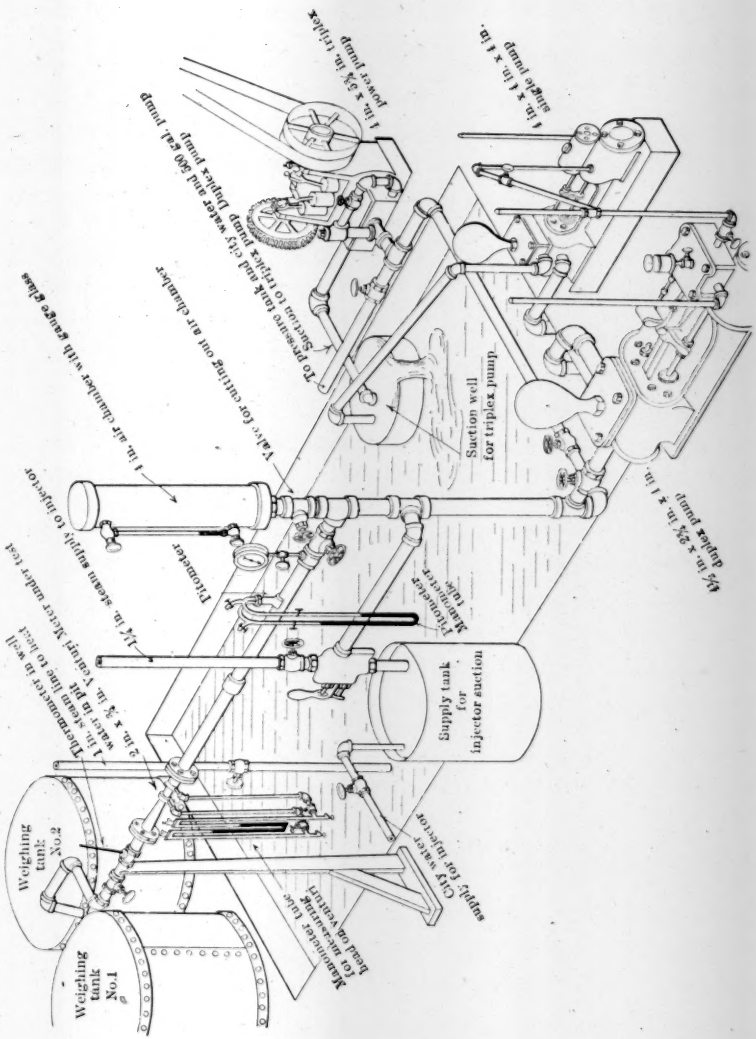
Weighing tank No. 2
Weighing tank No. 1
Thermometer in well
1 in. steam line to heater
2 in. × ¾ in. Vought Meter under test
¼ in. steam supply to injector
Manometer for measuring head on Venturi
Manometer tube
City water supply for injector
Supply tank for injector suction
Thermometer
Manometer tube
Pressure
1 in. air chamber with gauge glass
Valve for cutting out air chamber
To pressure tank and city water and 500 gal. pump
Suction to triplex pump
Duplex pump
Suction well for triplex pump
1 in. × 2¾ in. triplex power pump
1 in. × 1 in. triplex pump
6½ in. × 2¾ in. × 1 in. duplex pump

FIG. 1 LAYOUT OF APPARATUS, VENTURI METER TESTS

conditions, it was decided to determine the coefficients for this meter under these conditions. The discharge of the Venturi meter was figured from the regular formula, using a coefficient of one, and the actual weight obtained from the weighing tanks was divided by this value to obtain the real coefficient.

8 The following temperatures were used during the tests: 80 deg., 120 deg., 140 deg., and 180 deg. fahr. Water for these tests was supplied by the triplex power pump with different velocities through the meter. Tests were made at 140 deg. with and without the air chamber, the water being furnished by the triplex power pump; then at 140 deg. with one plunger of the triplex pump disconnected so as to produce fluctuations in the velocity and pressure of the water supplied to the meter.

9 In order to duplicate more nearly the conditions of boiler feed, an air chamber and check valve were placed in the pipe line in the downstream side of the Venturi meter (not shown in Fig. 1). Because of the air chamber at this end the pump fluctuations could pass through the meter to a much more marked degree than if the discharge was merely throttled by a valve. Under these conditions tests were run with water supplied by the injector, taking suction from a special supply tank; with the injector, however, the temperature of the water of necessity varied with the velocity through the meter.

CONCLUSIONS

10 The chief difficulty encountered in making this series of tests was in getting the true average readings of the manometer. With the higher velocities through the meter, the fluctuations could be easily dampened by closing the valves to the manometer tube, but with the lower velocities any error in reading was so large in proportion to the entire head as to make a considerable difference in results. It may be said, then, that the meter is not accurate for velocities of less than 10 ft. per sec. in the throat of the meter, which corresponds to a discharge of 0.03 cu. ft. per sec., or about 6140 lb. per hr., so that this meter would be best adapted for measuring water for a boiler plant of above 200 h.p. The coefficients are materially lower below 0.03 cu. ft. per sec. The principal feature shown by the cooler water tests (80 deg. fahr.) is the low value of the coefficients of the meter, the average, excepting the values for velocities below 10 ft. per sec., being 0.942. This coefficient might have been expected, however, as in a small meter the ratio of area of cross section to surface is much lower.

11 These experiments clearly show that the meter is as accurate for hot water as for cold. The maximum error in discharge, as figured from manometer deflections using the mean coefficients for that temperature, is as follows: 80 deg., 1.39 per cent; 120 deg., 1.5 per cent; 140 deg., 1.9 per cent; 180 deg., 0.82 per cent. The average error is well within 1 per cent.

12 Of the pumps tried with the meter, the triplex gave by far the best results, and it may be confidently stated that the Venturi feedwater meter would give very satisfactory results in a plant using the power pump. Even with one plunger disconnected, the maximum variation was only 2.4 per cent.

13 In tests with the injector, the weighed calculation from the Venturi formula, using the mean coefficient, shows variations from actual weight of 3 per cent. The average error is inside 2 per cent.

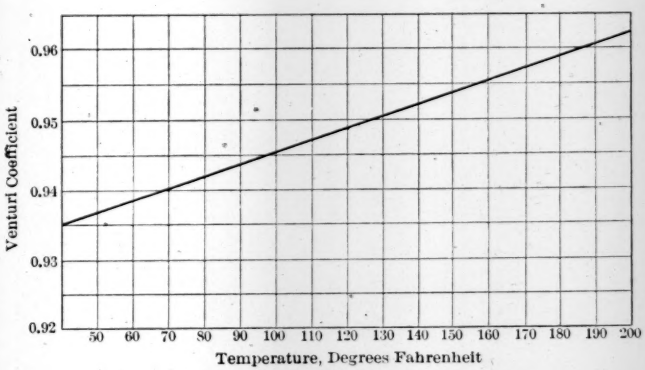

FIG. 2 CURVE SHOWING VARIATION OF VENTURI COEFFICIENT WITH RISE IN TEMPERATURE

14 The results from $4\frac{1}{2}$ in. by $2\frac{3}{4}$ in. by 4 in. duplex pump show up better than might be expected as this pump was in very poor condition, the piston rods being so worn that the pump took air through the stuffing boxes at the water end.

15 These tests represent the worst conditions which would be met with in boiler feed. The pump would start building up pressure, then pause until the pressure had fallen. The check valve, which was located only about 6 in. downstream of the Venturi meter, was opening and closing constantly. For a discharge of more than 0.03 to 0.04 cu. ft. per sec., the coefficients were very consistent.

16 It must be remembered in considering these tests that the Ven-

turi meter itself probably worked more accurately than the tests would indicate, as every change in velocity through the meter is accompanied by its corresponding change in head on throat and upstream end, and that a continuous recording device attached to the meter would probably cut down the error considerably. It is the opinion of the experimenters that the Venturi meter is a very reliable form of hot-water meter, provided the proper size is used.

17 These tests were conducted for the most part by George Y. Lancaster, a post-graduate student in mechanical engineering at Worcester Polytechnic Institute, and these results are taken from a thesis submitted by him.

THE DESIGN OF CURVED MACHINE MEMBERS UNDER ECCENTRIC LOAD

By Prof. Walter Rautenstrauch, New York
Member of the Society

Machine members, such as frames for punches, shears and riveters, hooks and the like, when subjected to load are generally supposed to behave like beams originally straight and subjected to the same conditions. The usual analysis applied to such beams in determining the proportions required to withstand safely a given stress assumes that the maximum tensile stress at a in Fig. 1 = load considered as uniformly distributed over the section + the stress due to the eccentricity of the load. Symbolically expressed

$$f_t = \frac{W}{A} + \frac{Wle}{I}$$

where

f_t = maximum intensity of tensile stress.
W = load on beam.
A = area of section.
I = eccentricity of loading.
e = distance from gravity axis of section to point under stress f_t.

2 This analysis is unfortunately prevalent in textbooks on the design of machine elements and strength of materials, and has been accepted generally because of long standing. However, it does not agree with the results of experiment on members of this kind; in fact such experimental results are so different from results calculated by this formula that no confidence whatever can be placed in it and safe proportions can be obtained only by the use of a large factor of safety.

3 The writer has recently published[1] the results of a series of

[1] American Machinist, October 7, 1909.

To be presented at a meeting of The American Society of Mechanical Engineers. All papers are subject to revision.

experiments which are remarkable in their disagreement with the results obtained by the formula. The crane hook was taken as an example of a beam of this sort and experiments were conducted on ten hooks ranging from two to thirty tons rated capacity. All hooks were furnished by the manufacturers. In Table 1 the results of the experiments are compared with the results by formula.

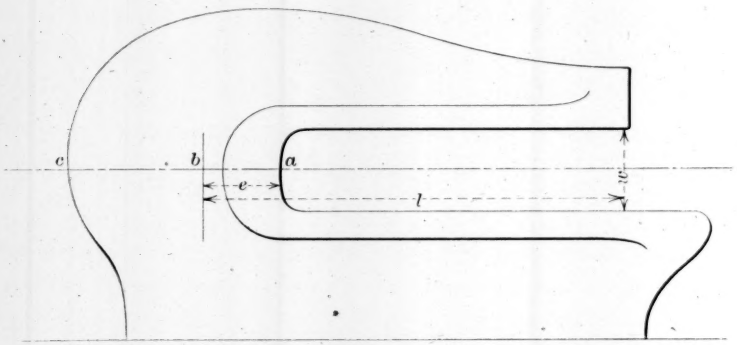

FIG. 1 MACHINE-TOOL FRAME CONSIDERED AS A LOADED BEAM

4 It is very evident that the assumptions on which the above formula is based are not correct, and that machine members designed on this basis have a much smaller factor of safety than is generally supposed. While this has been known in some quarters and attempts have been made to bring about an adjustment, no theory which has been developed seems to fit the case better than that evolved by E.

TABLE 1 COMPARISON OF RESULTS

DESCRIPTION OF HOOK	LOAD AT ELASTIC LIMIT BY TEST POUNDS	LOAD AT ELASTIC LIMIT BY STANDARD FORMULA POUNDS
30-ton cast steel	56,000	115,000
20-ton cast steel	30,000	70,000
15-ton cast steel	48,000	145,000
15-ton wrought iron	16,000	73,000
10-ton cast steel	18,000	43,000
10-ton wrought iron	16,000	26,000
5-ton cast steel	18,000	52,300
5-ton wrought iron	14,000	20,800
3-ton cast steel	8,500	14,900
2-ton cast steel	4,700	14,900

S. Andrews and Prof. Karl Pearson of London University. This analysis[1] is deserving of much more attention than it has received, and it is surprising that even some who have had access to it have made the statement that the old theory was sufficiently accurate for the usual case of design. It will not be my purpose to give a complete derivation of the new formula, which has been published elsewhere[1] in complete detail, but rather to show how the results of the analysis may be made directly applicable to design.

5 The investigation referred to gives the following expression for the tensile stress at the most strained point in the principal section of beam:

$$f_t = \frac{W}{A}\left\{\frac{l}{\rho\gamma_2}\left(\frac{1}{\left(1-\frac{e}{\rho}\right)^{\frac{1}{2}}}-\gamma_1\right)+1\right\}$$

where

f_t = tensile stress at most strained point of section, pounds per square inch.

W = load on hook, pounds.

A = area of section, square inches.

l = distance from load line to gravity axis of section.

ρ = radius of curvature of belly of hook at gravity axis.

e = distance from gravity axis to point of maximum tensile stress.

γ_1 = a function whose value is equal to $\Sigma \dfrac{dA}{\left(1+\frac{y}{\rho}\right)^{\frac{1}{2}}}$ in which A

dA is any differential area of section, a distance y from the gravity axis, Σ denoting the sum of all the operations indicated by the symbols.

γ_2 = a function whose value is equal to $\gamma_1 - \Sigma \dfrac{dA}{\left(1+\frac{y}{\rho}\right)^{\frac{1}{2}}}$ A

6 The functions γ_1 and γ_2 are to be found for any section, as shown in Fig. 2. The half-area of the hook is *akhlc*. At any distance y

[1]Technical Series 1, Draper Company's Research Memoirs, 1904.

from b erect an ordinate equal to $\frac{d\,g}{\left(1+\frac{y}{\rho}\right)^{\frac{1}{4}}}$. Such an ordinate is de. All possible ordinates of this nature determine the curve $aehjc$. Clearly then

$$\gamma_1 = \frac{\text{area } aehjc}{\frac{1}{2}\text{ area of section}}$$

Likewise, if at any distance y from b an ordinate is erected whose

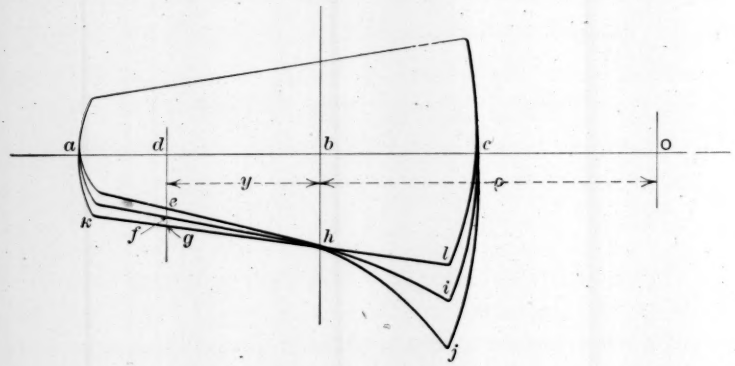

FIG. 2 CROSS-SECTION OF A HOOK

length is $\frac{d\,g}{\left(1+\frac{y}{\rho}\right)^{\frac{1}{4}}} = df$, all possible ordinates of this nature form the curve $afhic$, and $\gamma_2 = \frac{\text{area } aehjc - \text{area } afhic}{\frac{1}{2}\text{ area of section}}$.

7 The above analysis was applied to each of the hooks tested, with the results recorded in Column 3 of Table 2. An inspection of

TABLE 2 ANALYSIS OF HOOKS TESTED

LOAD AT ELASTIC LIMIT, POUNDS

RATED CAPACITY	30-TON CAST STEEL	20-TON CAST STEEL	15-TON		10-TON		5-TON		3-TON	2-TON
			Cast Steel	Wrought Iron	Cast Steel	Wrought Iron	Cast Steel	Wrought Iron	Cast Steel	Cast Steel
By test	56,000	30,000	48,000	16,000	18,000	16,000	18,000	14,000	8,500	4,700
By standard formula	115,000	70,000	145,000	73,000	43,000	26,000	52,300	20,800	14,900	14,900
By new formula	55,080	29,925	50,570	15,000	16,500	15,000	18,950	14,100	8,600	4,400

this table will show how nearly the analysis of Mr. Andrews and Professor Pearson fits the case and how far from correct are the results from the old formula. The new formula appears then to be based on a correct theory and to be perfectly safe for use in the design of all machine members of this general type.

8 In its present form it is a rather unwieldy instrument in the hands of a designer, but it may be made more applicable to design than might be thought at first. Upon examination it will be seen that the functions γ_1 and γ_2 are constants for all sections of similar form, that is, for all sections the proportions of which may be expressed as a function of some unit of dimension, for example, the radius of curvature. Under the same circumstances the entire expression within the brackets is a constant. The equation for a series of sizes and sections may therefore be written $f_t = \frac{W}{A}K$, or $A = \frac{W}{f_t}K$. The area is a function of the unit squared and therefore we may write $A = C'r^2$, or

$$r = \sqrt{\frac{K}{C'}\frac{W}{f_t}} = C\sqrt{\frac{W}{f_t}}$$

Applying this to the case of a series of hooks ranging from the minimum to the maximum to be manufactured, a standard form of section may be laid out as in Fig. 3, and the constant established. For the hooks tested by the writer the following values for the constant were found:

30-ton hook, cast steel	3.00
30-ton hook, cast steel	3.10
15-ton hook, cast steel	3.23
15-ton hook, wrought iron	4.29
10-ton hook, cast steel	3.49
10-ton hook, wrought iron	3.42
5-ton hook, cast steel	3.12
5-ton hook, wrought iron	3.12
3-ton hook, cast steel	3.78
2-ton hook cast steel	3.74
Average	3.43

9 To make the case representative of present practice let such ratio of proportions be assigned to the section shown in Fig. 3 that $C = 3.4$. The design of a series of wrought iron hooks to sustain

loads of from 2 to 40 tons with a limiting intensity of tensile stress of 30,000 lb. per sq. in. will require the following computations:

$$\text{40-ton hook, } r = 3.4\sqrt{\frac{80000}{30000}} = 5.54$$

$$\text{30-ton hook, } r = 3.4\sqrt{\frac{60000}{30000}} = 4.7$$

$$\text{20-ton hook, } r = 3.4\sqrt{\frac{40000}{30000}} = 3.94$$

$$\text{10-ton hook, } r = 3.4\sqrt{\frac{20000}{30000}} = 2.76$$

$$\text{5-ton hook, } r = 3.4\sqrt{\frac{10000}{30000}} = 1.95$$

$$\text{2-ton hook, } r = 3.4\sqrt{\frac{4000}{30000}} = 1.23$$

10 The proportions obtained above will be for loads giving a maximum stress at the elastic limit of the material. For cast steel differ-

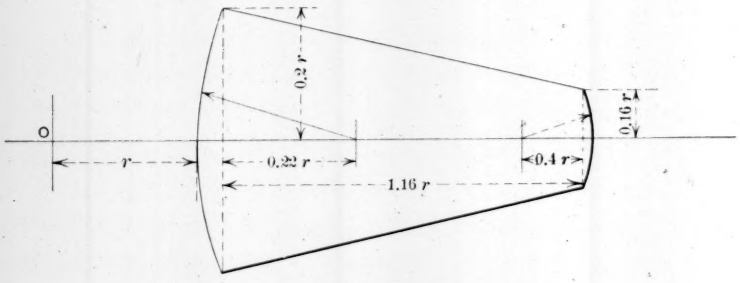

FIG. 3 STANDARD HOOK SECTION

ent values will necessarily be obtained. The establishment of such a standard would lead to a very simple process for the determination of the principal section of a hook for any capacity; the proportions of the shank and other parts of the hook may readily be established on the same basis. The bottom of the hook, being subjected to much wear, cannot of course be proportioned on the basis of the stress analysis. The above standard section selected as an average representative of present practice is not, however, the most economic form of section from the standpoint of equal maximum tensile and compressive stresses. It has been pointed out by Professor Pearson that a

section with such proportions is approximately an isosceles triangle with a radius of curvature of 1.75 of the height. The more nearly this form could be approached the less would be the weight of hook for the same capacity.

11 Professor Goodman[1] points out that for hook sections the functions γ_1 and γ_2 are expressed approximately as follows:

$$\gamma_2 = \frac{ke}{1.2\,\rho^2}$$

$$\gamma_1 = 1 + 1.1\,\gamma_2$$

where k = radius of gyration of the sections, the other symbols being as before noted.

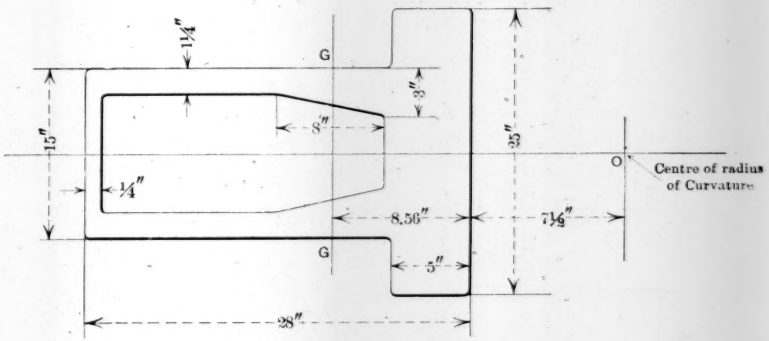

FIG. 4 DESIGN FOR A PUNCH FRAME

$A = 212$ $I = 14{,}533$ $k = 68.56$ $l = 38.56$ $\rho = 16$ $e = 8.56$

12 It will be found much more convenient to use these empirical expressions, which give quite accurate results, than to determine the value of the functions by the more tedious graphic method.

13 In applying these empirical formulæ to punch and riveter frame sections the writer has found that the results are not accurate but that the values are better expressed as follows:

$$\gamma_2 = \frac{ke}{0.7\,\rho}$$

$$\gamma_1 = 1 + 1.1\,\gamma_2$$

For example, consider the design for a punch frame shown in Fig. 4

[1]Institute of Mechanical Engineers, Proceedings, vol. 167.

Computing the values for the functions γ_1 and γ_2 by the graphic method, $\gamma_1 = 1.4$, $\gamma_2 = 0.405$. Whereupon the intensity of stress according to the new method of analysis for a force of 90,000 lb. at the punch will be

$$f_t = \frac{W}{A}\left\{\frac{l}{\rho\gamma_2}\left(\frac{1}{\left(1-\frac{e}{\rho}\right)}-\gamma_1^{\frac{5}{4}}\right)+1\right\} = 8500 \text{ lb. per sq. in.}$$

The values of γ_1 and γ_2 by the empirical formula are 1.44 and 0.4 respectively. Whereupon the intensity of stress becomes $f_t = 8500$ approximately. According to the old formula used almost exclusively in textbooks, the value of f_t is expressed by $\frac{W}{A}+\frac{Wle}{I}$, whence $f_t = 2450$.

14 The above empirical formulæ are derived from the results of computation of two sections. I am not prepared to state that it will work out in all cases and must therefore caution anyone against using these values to check the results by the graphic method. It may be clearly seen that were the punch in question designed for a limiting intensity of stress of 2450 by the old formula, there would actually be a maximum stress of 8500 lb. per sq. in., which is hardly a safe value for cast iron and particularly for a large casting.

COMMENT ON CURRENT BOOKS

REINFORCED CONCRETE IN EUROPE. By Albert Ladd Colby. *The Chemical Publishing Co. Easton, Pa.* 1909. Cloth, 6 by 9; x + 260 p. Price $3.

This book contains a report made to a number of "Subscribers" on current practice in reinforced concrete construction in Great Britain and on the continent, the material being collected in 1908, chiefly by personal interviews with the leading authorities in each country. Theory is but briefly dealt with, while the practical side is fully discussed, including the economy and proof of the endurance of reinforced concrete construction, the various systems in use, specifications, ingredients, mixing and construction. Considerable space is given up to the kind of steel and forms of bars in use abroad. To enable the reader to obtain further information the addresses of prominent consulting and contracting engineers of each country are given. An appendix gives the addresses of official and technical testing stations, congresses, institutions, and the like. A bibliography is also given of books, journals and periodicals dealing with the subject in each country.

Contents by chapter headings: Applications of Reinforced Concrete; Economies of Reinforced Concrete Construction; Endurance of Foreign Reinforced Concrete Construction; Foreign Systems of Reinforced Concrete Construction; Mechanical Bond and Forms of Bars; Metal Used for Reinforcement; Foreign Specifications, Recommendations and Opinions Compared; Cement Used in Reinforced Concrete; The Chief Requirements of Foreign Cement Specifications Compared; Concrete Used in Reinforced Concrete; The Chief Requirements of Foreign Concrete Specifications Compared; Reinforced Concrete; Foreign Specifications and Recommendations Compared under the Chief Specified Requirements; Lists and Description of Foreign Government and Private Testing Stations, Congresses, Technical Institutions, Associations, and Committees, who have Adopted Resolutions, Specifications, or Rules Relating Thereto; Bibliography on Reinforced Concrete, Concrete and Cement.

THE RAILWAY LOCOMOTIVE. By Vaughan Pendred. *D. Van Nostrand Co. New York.* 1908. Cloth, 6 by 8; ix + 310 p.; 94 illustrations. Price $2.

The author's reason for writing this book is that he wished to deal with locomotives not "anatomically," but "physiologically." "The technical treatise deals with the locomotive almost altogether as a machine. Its parts are described, but the reasons why they assume particular shapes, and why one shape is better or worse than another are not dwelt upon, and nothing is said about the daily life of the engine." The author therefore endeavors to tell why the locomotive "is what it is." He explains the mechanical and physical phenomena on which it depends for its action, and the objects kept in view by those who design, construct and use it. He has treated the locomotive from the three standpoints of vehicle, steam generator and steam engine. The book is not intended as a "popular" work but is for engineers young in the profession. Though British practice is mainly considered, Europe and the United States receive considerable attention. The method of presentation is clear, concise and interesting.

Contents by chapter headings: Frames; Bogies; The Action of the Bogie; Center of Gravity; Wheels; Wheel and Rail; Adhesion; Propulsion; Counter-Balancing; The Boiler; The Construction of the Boiler; Stay Bolts; The Fire Box; The Design of Boilers; Combustion; Fuel; The

Front End; The Blast Pipe; Steam; Water; Priming; The Quality of Steam; Superheating; Boiler Fittings; The Injector; Cylinders and Valves; Friction; Valve Gear; Expansion; The Stephenson Link Motion; Walschaert's and Joy's Gears; Slide Valves; Compounding; Piston Valves; The Indicator; Tenders; Tank Engines; Lubrication; Brakes; The Running Shed; The Work of the Locomotive.

METALLURGY OF THE COMMON METALS (NON-FERROUS). By A. Humboldt Sexton and John S. G. Primrose. *The Scientific Publishing Co. Manchester, England.* Cloth 6 by 9; xvii + 483 pages; 185 illustrations. Price 7s. 6d.

The need felt by the authors in their work in the Glasgow and West of Scotland Technical College for a text book of the metallurgy of the common (non-ferrous) metals, has prompted the preparation of the present book. It is intended chiefly as an introduction to further study, to pursue which the student is referred to a number of books listed in an appendix.

Contents by chapter headings: Properties of Copper; Copper Minerals; The Principles of Copper Smelting; Copper Processes; Ore Dressing; Modern Methods of Copper Smelting; Matte Smelting in Blast Furnaces; Coarse Copper in Reverberatory Furnaces; Copper Refining; Wet or Leaching Processes; Electro-deposition of Copper; Properties of Tin; Working Alluvial Deposits; Tin Smelting in Reverberatory and Blast Furnaces; Refining Tin; Properties of Zinc; Dressing Zinc Ores; The English, Belgian, Silesian, and Rhenish Processes; Effect of Impurities; Properties of Lead; Preparation of Lead Ores; Smelting in Hearths; Roasting Lead Ores; Blast Furnace Smelting; Blast Furnaces for Lead Smelting; Products of Lead Smelting Fume Recovery; Wet and Electrolytic Processes; Softening and Refining Lead; Concentration of Silver in Lead; Cupellation; Electrolytic Refining of Lead; Antimony; Properties of Aluminum; Process for the Preparation of Aluminum; Properties of Nickel; Dry Methods for the Extraction of Nickel, Wet and Electrolytic Methods; Physical Constants of Metals; Thermo-Chemical Data; Factors in Calculating Blast Furnace Charges; Bibliography.

THE ELEMENTS OF MECHANICS OF MATERIALS. By C. E. Houghton. *D. Van Nostrand Co. New York.* 1909. Cloth 6 by 8; viii + 186 p.; 88 illustrations. Price $2.

The author has prepared this volume for use as an elementary text book. The extreme mathematical treatment is avoided, but where higher mathematics leads to clearness it is freely used. Review questions are given at the end of each chapter. The notation is uniform with that of Merriman's works, so that his more complete treatise may be used as a reference book.

Contents by chapter headings: Applied Mechanics; Applications; Beams; Torsion; The Elastic Curve; Long Columns; Combined Stresses; Compound Bars and Beams.

MECHANICAL ENGINEERING AND SHOP PRACTICE. By Stanley H. Moore. *Hill Publishing Co., New York,* 1908. Flexible Leather, 8vo., xv + 502 p., illustrated. Price, $4.

About one-half of this volume is devoted to a treatment of the ordinary machine shop processes and the use of machine tools, as indicated in the table of contents below. The balance of the book contains related matter of a broader character not usually included in books on shop work, but upon which the skilled mechanic should be informed in order to accomplish work intelligently. The matter referred to includes a chapter on materials, giving the strength, composition and other characteristics besides methods of hardening steel. Discussion of friction and lubrication, an outline of the principles of cutting tools, and data upon measuring and gaging work and upon screw threads, follow. The book is concluded with chapters upon power-generating machines, power transmission,

motor driving, etc. The author has included his paper delivered before this Society upon Force and Running Fits.

Contents by chapter headings: Introduction and Equipment; Materials; Friction, Lubricants and Lubrication; Cutting Tools; Measuring and Small Tools; Screw and Pin Data; Bench and Vice Work; Turning; Boring; Drilling; Grinding; Planing; Milling; Miscellaneous Machine Tools and Accessories; Shop Processes and Kinks; Mechanics; Power-Generating Machines; Elementary Electricity; Power Transmission; Motor Drives and Motor-Driven Machine Tools.

PROCEEDINGS OF A CONFERENCE OF GOVERNORS IN THE WHITE HOUSE, Washington, D. C., May 13–15, 1908. Issued under the direction of the Committee of Governors, by W J McGee, Recording Secretary. *Washington, Govt.* Cloth, xxxvii + 451 p., illustrated.

This is a verbatim record of the six sessions of the Conference of Governors called by the President of the United States to consider the conservation of the national resources. At the conference M. L. Holman, President and Calvin W. Rice, Secretary, represented The American Society of Mechanical Engineers, by invitation of President Roosevelt, and the Secretary attended also as a specially invited guest. The discussion of the Secretary is published on Page 420, and that of H. G. Stott, Member of Council, Am. Soc.M.E., but representing the American Institute of Electrical Engineers in his capacity of President, is given on Page 404.

The *contents* include addresses and discussion on The Conservation of Ores and Related Minerals, The Waste of Our Fuel Resources, The Natural Wealth of the Land and its Conservation, Soil Wastage, Forest Conservation, Resources Related to Irrigation, Grazing on the Public Lands, Conservation of Life and Health by Improved Water Supply, Navigation Resources of American Waterways, Conservation of Natural Resources in Illinois, New York, Pennsylvania, Michigan, Wyoming, Hawaii, Washington, D. C., Florida, Iowa, Maryland, Arizona, Maine Washington, Conservation of Human Life, Our Water Resources, The Lakes-to-Gulf Waterway, Plans for Conservation, Forestry as Related to Mining Interests, The Conservation Problem, Conservation from the View-point of Recreation, Water Resources, Methods for Conservation, The Immediate Necessity for Acquiring the Appalachian Forest Reserve, Forest Conservation, Suggestions on the Conservation of Coal, Suggestions on the Conservation of Some of our Resources, Conservation in Relation to Labor, View of the Engineer, Conservation of Minerals, Railways and Conservation, The Preservation of Scenic Beauty, The Use of Some of the Natural Resources of the Country and Possible Economies of their Use, Interests of the Manufacturer, Conservation of Soils, The Twilight Zone, Necessity for Waterway Improvement, Fire Prevention, How Conservation of Mineral Resources can be Accomplished.

TABLES OF THE PROPERTIES OF STEAM AND OTHER VAPORS AND TEMPERATURE-ENTROPY TABLE. For Engineers and Students. By Cecil H. Peabody. *John Wiley & Sons, New York,* 1909. Eighth edition, rewritten. Cloth 8vo, v + 133 p. Price $1.

Peabody's tables of properties of saturated steam were published over twenty years ago and are in general use by engineers. In the intervening time important investigations upon the properties of steam have been made by various physicists. With these new data as a basis the original tables were recomputed and extended temperature-entropy tables were added. The edition containing the latter was issued in 1907. An edition has now been brought out in which the tables have again been recalculated to embody the refinements now rendered possible by recent important determinations relating to the properties of steam. The temperature-entropy tables give the quality, heat contents and specific volume of saturated and superheated steam for each degree of temperature fahrenheit; and for each hundredth of a unit of entropy. Such tables have become invaluable since the advent of the steam turbine in which the action of steam is assumed to

be approximately adiabatic. Calculations upon the adiabatic flow of steam and upon conditions existing in the different stages of a turbine are readily made by the use of such tables without the tedious mathematical work otherwise necessary.

Contents: Introduction; Tables; Steam, English Units, Degrees; Steam, English Units Pounds; Steam, French and English Units; Ether; Alcohol; Chloroform; Carbon Bisulphide; Carbon Tetrachloride; Aceton; Ammonia; Sulphur Dioxid; Specific Volumes of Liquids; Volumes of Hot Water; Inches of Mercury and Pounds; Corrective Factors, Superheated Steam; Temperature-Entropy Table; Naperian Logarithms; Four-Place Logarithms.

SYSTEMS OF GOVERNING AND VALVE-GEARS OF EUROPEAN GAS ENGINES. By R. E. Mathot, Mem.Am.Soc.M.E. For sale by the author. *Brussels (Belgium)*, 1908. Pamphlet form, 58 pages, illustrated. Price 3s.

This is a comprehensive discussion of valve gears and methods, and devices used for governing gas engines of both large and small sizes. The mechanisms illustrated are grouped under classifications relating to the principles underlying the different systems of operation; such as Governing at Constant Ratio and Volume, Variable Ratio and Constant Volume, etc. The whole comprises a systematic study of the subject.

GEAR-CUTTING MACHINERY. By Ralph E. Flanders, Assoc.Am.Soc.M.E. *John Wiley & Sons, New York*, 1909. First edition. Large 12mo, viii + 319 p., 219 illustrations. Price $3.

The author of this book contributed the paper at the last annual meeting of the Society upon Interchangeable Involute Gear Tooth Systems, which resulted in the appointment of a committee to formulate gear-tooth standards. His treatise upon gear-cutting machinery is unique in that it treats solely and with thoroughness of the different types of one form of machine tool. It explains the underlying principles of the different systems of gear cutting in use at the present time and of the mechanisms by which these systems are carried out in modern gear-cutting machines. As far as possible all of the European and English machines as well as American, are included. The method of treatment is to describe the machines and their mechanisms in the order determined, first, by the form of gear each is designed to cut; next, by the principle of action involved; next, by the method of operation employed; then by the kind of mechanism used; and finally, by the structural design of the machine. Most of the material comprising the book has appeared serially in *Machinery* of which its author is associate editor.

Contents by chapter headings: Methods of Forming the Teeth of Gears; Machines for Forming the Teeth of Spur Gears; Machines for Cutting the Teeth of Internal Gears and Racks; Machines for Cutting the Teeth of Worms and Helical Gears; Worm-Wheel Cutting Machines; Machines for Forming the Teeth of Bevel Gears.

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WILBUR R. KIMBALL, *New York City.* Announcement of evening course on Aëronautics, to be given in West Side Y. M. C. A. by Mr. Kimball.

JOHN H. MORRISON, *Brooklyn, N. Y.* Announcement of "History of New York Ship Yards," by Mr. Morrison, showing the development of wooden ship building in the United States.

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EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 15th of the month. The list of men available is made up of members of the Society. and these are on file, with the names of other good men not members of the Society who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

074 Capable, energetic young man, not less than thirty years of age, to handle correspondence. Must have good business judgment. Supply all details and state salary expected. Location, Detroit, Mich.

075 Master mechanic for a beet sugar factory in Southern California employing about sixty men during repair seasons. Man must be capable of handling labor, and have had a varied experience in general machinery work. State age, past experience in full, and salary expected.

076 Mechanical draftsman, recent graduate in mechanical engineering, capable of designing pumps and heavy hoisting machinery, experienced in use of steam-engine indicator, and able to assist in testing all sorts of mechanical equipment, and making computations.

077 First-class structural draftsman, able to carry through complete designs for head-frames, tipples, coal-pockets and all sorts of buildings. Give full particulars as to experience, age and salary expected.

078 Position available for competent designer of steam pump machinery; the position would be that of assistant to the chief designer. Familiarity with hoisting machinery would be an asset. Location Middle West.

MEN AVAILABLE

311 Cornell graduate, electrical and mechanical engineer, experience as chief engineer on the design and construction of large steam-turbine power and centrifugal pumping plants, distribution systems, fire-protection systems and the electrification of industrial plants, desires to make engagement on the completion of present work, about January 1.

312 Graduate mechanical and electrical courses, W. P. I., age thirty-one, desires position in engineering or executive capacity. Experienced in engineering contracting business, and construction; has installed, repaired and operated various types of gas, steam and electrical power equipments. Competent to

prepare plans, specifications, estimates and reports; six years on the Pacific coast and previously in New England. Salary $2500. Location immaterial.

313 Experienced mechanical and electrical engineer, technical graduate, wishes responsible position with a first-class firm of consulting engineers. Is thoroughly experienced in economical operation of power plants, having served for a number of years as operating engineer in charge of a group of stations. Conversant with the most recent practice in design and operation.

314 Technical graduate, age thirty-three, active, some years' practical experience, employed and had charge of man, desires position as assistant to superintendent, testing engineer or similar position.

315 Member desires position as factory manager or mechanical engineer with concern manufacturing light or medium weight work. Long experience, best references.

316 Member, with wide commercial and technical experience in operation, design, construction and laying out of industrial plants , is open for engagement November 1.

CHANGES IN MEMBERSHIP

CHANGES OF ADDRESS

ADAMS, Kilburn E. (Junior, 1908), Mech. Engr., Boston & Albany R.R., Rm. 372, South Sta., Boston, and for *mail*, 132 Oxford St., North Cambridge, Mass.

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ALLEN, Walter C. (Junior, 1905), Genl. Mgr., Yale & Towne Mfg. Co., 9 Murray St., New York, N. Y., and 181 Greylock Pl., Stamford, Conn.

ANDERSON, Harry Warfield (Associate, 1907), 103 Candler Bldg., and *for mail*, Owens Apts., 5 E. 3d St., Atlanta, Ga.

BELL, John Everett (Associate, 1903), Mech. Engr., Babcock & Wilcox Co., 85 Liberty St., New York, N. Y., and Oradell, N. J.

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CHAMBERS, Norman C. (Junior, 1905), 136 W. 44th St., New York, N. Y.

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FRANCIS, W. H. (1884), Union League Club, Philadelphia, Pa.

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HUMPHREYS, Alex. C. (1884), Manager, 1907–1910; Life Member; Pres., Stevens Inst. of Tech., Hoboken, N. J., Pres., Buffalo Gas Co., and *for mail*, Pres., Humphreys & Glasgow, 165 Broadway, New York, N. Y.

KUNZE, Edward J. (1907), Instr. Steam and Gas Engrg., Univ. of Wis., Madison, Wis.

LOCKWOOD, James Fred (1889; 1907), Mgr., Security Elev. Safety Co., 126 W. 18th St., New York, and *for mail*, 678 McDonough St., Brooklyn, N. Y.

McELROY, Jos. A. (1895), McElroy Bros., Bridgeport, Conn.

MacFARLAND, Helon Brooks (1907), Engr. of Tests, Atchison, Topeka & Santa Fe Ry. Co., Topeka, Kan.

MAHL, Fred W. (Junior, 1892), Union Pacific System and Southern Pacific Co., 135 Adams St., Chicago, Ill.

MORGAN, John R. (1901), 10227 S. Wood St., Chicago, Ill.

MURRAY, Henry H. (1899; 1905), Ch. Draftsman, Victor Talking Mch. Co., Camden, and *for mail*, 406 Lippincott Ave., Riverton, N. J.

MYERS, Curtis Clark (Junior, 1905), Shop Coördinator, Univ. of Cincinnati, Cincinnati, O.

O'KEEFE, James G. (Junior, 1907), Cadet Engr., Pub. Service Corp. of N. J., and *for mail*, 925 S. 16th St., Newark, N. J.

OLIVER, E. C. (Junior, 1902), 519 Cass Ave., Detroit, Mich.

PINNER, Seymour Wm. (Junior, 1909), Instr., Univ. of Mich., Ann Arbor, Mich.

RHOADS, George Elwood (Junior, 1906), Inspr. Tests Dept., Penn. R. R., 12th Ave. and 12th St., and *for mail*, 1320 20th Ave., Altoona, Pa.

ROBINSON, Edward (1891; 1902), Prof. of Mech. Engrg., Univ. of Vt., and *for mail*, 25 Colchester Ave., Burlington, Vt.

ROGERS, Robert W. (Junior, 1908), Erie R. R., and *for mail*, 216 Walnut St., Meadville, Pa.

ROSE, William Holladay (Junior, 1901), present address unknown.

ROWLEY, Harry Wm. (1896), Allis-Chalmers Co., 912 Evans Bldg., Washington.

SAR VANT, Wilbur Nason (Junior, 1907), 16 Verona Pl., Brooklyn, N. Y.

SERGEANT, Chas. H. (1895), 525 W. 138th St., New York, N. Y.

SLAUSON, Harold W. (Junior, 1908), Asso. Tech. Press Bureau, 25 W. 42d St., Lock Box 27, Times Sq. Sta., New York, and *for mail*, 208 Bay 22d St., Brooklyn, N. Y.

SMITH, William E. (Junior, 1908), Babcock & Wilcox Co., Barberton, O.

SPEER, Chas. Henry (Associate, 1907), Asst. Engr., Algoma Steel Co. Ltd., Sault Ste. Marie, Ontario, Canada.

STOUGHTON, Edwin R. (Associate, 1907), Baird & West, 149 Jefferson Ave., Detroit, Mich.

TAGGE, Arthur C. (1901), Engr., Internatl. Portland Cement Co., Ottawa, Ont.

THOMAS, Jay G. (Junior, 1906), Draftsman, Westinghouse Mch. Co., E. Pittsburg, and *for mail*, 500 Ellicott St., Wilkinsburg, Pa.

VAN DEINSE, A. F. (Junior, 1905), Westinghouse Elec. Co., and *for mail*, 123 San Francisco St., El Paso, Texas.

VINCENT, Jesse Gurney (Associate, 1908), Supt. of Inventions, Burroughs Adding Mch. Co., and *for mail*, 301 Addison Apts., Detroit, Mich.

VON GOEBEN, Carl (1899), Engr. and Contr., 140 Cedar St., New York, N. Y., and *for mail*, 34 Irving St., Montclair, N. J.

WATSON, Herbert L. (Junior, 1907), Sales Engr., Allis-Chalmers Co., 50 Congress St., Boston, Mass.

WIDDICOMBE, R. A. (1898; 1906), V. P. and Mgr., Dixon Steam System Co., First Natl. Bank Bldg., and *for mail*, 5552 Lakewood Ave., Chicago, Ill.

DEATHS

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McGWIRE, Charles H. (1909), U. S. P. O., Atlanta, Ga.

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ANDERSON, Harry Warfield (1909), 103 Candler Bldg., and *for mail*, Owens Apts., 5 E. 3d St., Atlanta, Ga.

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ROWLEY, Harry Wm. (1909), Allis-Chalmers Co., 912 Evans Bldg., Washington, D. C.

NEW MEMBERS

HAMPTON, Wm. M. (Affiliate, 1909), Supt. Gas Engs., Indiana Steel Co., and *far mail*, P. O. Box 208, Gary, Ind.

MALLON, Michael F. (Affiliate, 1909), Ch. Engr., N. Y. Div., Swift & Co., 625 Brook Ave. and 604 Tinton Ave., New York, N. Y.

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WIGTEL, Carl (Affiliate, 1909), Ch. Engr., Watson-Stillman Co., 50 Church St., New York, N. Y.

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CHANGES OF ADDRESS

BRAGAW, Richard (Student, 1909), 838 Willett St., Jamaica, N. Y.

HIGGINS, Warren S. (Student, 1909), 9 Prospect Park W., Brooklyn, N. Y.

HOLLENBERGER, Theo. J. (Student, 1909), 4433 N. Paulina St., Chicago, Ill.

WOODWARD, George W. (Student. 1909), 494 Main St., Worcester, Mass.

COMING MEETINGS

OCTOBER AND NOVEMBER

Secretaries or members of societies whose meetings are of interest to engineers are invited to send in their notices for publication in this department. Such notices should be in the editor's hands by the 18th of the month preceding the meeting.

ALABAMA LIGHT AND TRACTION ASSOCIATION

November 15, 16, annual convention, Birmingham. Secy., Lloyd Lyon, 158 Government St., Mobile.

APPALACHIAN ENGINEERING ASSOCIATION

November 5, 6, Washington, D.C. Secy., H. M. Payne, Morgantown, W. Va.

AMERICAN CIVIC ASSOCIATION

November 15-19, Cincinnati, O. Secy., Richard B. Watrous, Harrisburg, Pa.

AMERICAN GAS INSTITUTE

October 22-20, Hotel Pontchartrain, Detroit, Mich. Secy., A. B. Beadle, 25 West 39th St., N. Y.

AMERICAN RAILWAY ASSOCIATION

November 17, annual meeting, Chicago, Ill. Secy., W. F. Allen, 24 Park Pl., New York.

AMERICAN RAILWAY BRIDGE AND BUILDING ASSOCIATION

October 19-21, annual convention, Jacksonville, Fla. Secy., S. F. Patterson, Boston & Maine Ry., Concord, N. H.

AMERICAN SOCIETY OF CIVIL ENGINEERS

October 20, Paper: Characteristics of Modern Hydraulic Turbines, Chester W. Larner. Secy., Chas. W. Hunt, 220 W. 57th St., New York.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

October 16, Engineers' Club, St. Louis, Mo.; October 20, Chipman Hall, Boston, Mass.: November 9, 29 W. 39th St., New York. December 7-10, annual meeting, New York. Secy., C. W. Rice, 29 W. 39th St., New York.

AMERICAN SOCIETY OF MUNICIPAL IMPROVEMENTS

November 9-11, annual meeting, Little Rock, Ark. Secy., A. Prescott Folwell, 239 W. 39th St., New York.

AMERICAN SOCIETY OF REFRIGERATING ENGINEERS

October 18, 19, Stratford Hotel, Chicago, Ill. Secy., W. H. Ross, 154 Nassau St., New York.

BROOKLYN ENGINEERS' CLUB

October 21, Paper: Reinforced Concrete as Applied to Piles and Cross Ties, Alex C. Chenoweth; October 28, The Reconstruction of the Hanson Place Sewer at Flatbush Ave.; November 4, Laboratory Tests of Materials for Coating Steel Pipe by Melville C. Whipple. Secy. Joseph Strachan, 117 Remsen St., Brooklyn, N. Y.

CENTRAL RAILWAY CLUB

November 12, Hotel Iroquois, Buffalo, N. Y., 8 p.m. Paper: Application of Electricity to the Movement of Freight, G. H. Condict. Secy., H. D. Vought.

CENTRAL RAILWAY AND ENGINEERING CLUB OF CANADA

October 19, November 16, Prince George Hotel, Toronto, Ont. Papers: Care and Maintenance of Elevators, J. Shales; Gas Engines, their Origin and Commercial Use, G. M. Henderson. Secy., C. L. Worth, Union Sta.

EMPIRE STATE GAS AND ELECTRIC ASSOCIATION

November 17, 18, 29 W. 39th St., New York. Secy., C. H. B. Chapin.

ENGINEERS' CLUB OF TORONTO

October 21, 8 p.m., 96 King St. W. Paper: The Air as a Medium of Transportation, J. F. d'Almeida. Secy., R. B. Wolsey, 25 Lowther Ave.

MODERN SCIENCE CLUB

October 19, 125 S. Elliott Pl., Brooklyn, N. Y. Paper: Air Compressors, S. B. Redfield. October 26. Paper, Power, F. L. Johnson. Secy., J. A. Donnelly.

NATIONAL MUNICIPAL LEAGUE

November 15–18, Cincinnati, O. Secy., C. R. Woodruff, 121 S. Broad St., Philadelphia, Pa.

NEW YORK RAILROAD CLUB

November 19, annual meeting, 29 W. 39th St. Secy., H. D. Vought, 95 Liberty St.

OHIO SOCIETY MECHANICAL, ELECTRICAL & STEAM ENGINEERS

November 19, 20, main annual meeting, Lima, O. Secy., David Gaehr, Schofield Bldg., Cleveland.

RAILWAY CLUB OF PITTSBURG

October 22, annual meeting, Monongahela House. Secy., J. D. McIlwain, care P. & L. E. R. R.

RICHMOND RAILROAD CLUB

November 8, annual meeting. Secy., F. O. Robinson.

SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS

November 18–19, annual meeting, 29 W. 39th St., New York. Secy., W. H. Baxter.

SOUTHERN AND SOUTHWESTERN RAILWAY CLUB

November 18, annual meeting, Candler Bldg., Atlanta, Ga. Papers on Oil Lamps, Front-End Arrangements, Draft-Rigging. Secy., A. J. Merrill, 218 Prudential Bldg.

WESTERN ASSOCIATION OF ELECTRICAL INSPECTORS

October 26–28, annual meeting, Detroit, Mich.

WESTERN SOCIETY OF ENGINEERS

October 20, November 3, 20, 1735 Monadnock Blk., Chicago, Ill. Papers: Hydraulic Mining of Auriferous Gravels, J. W. Phillips; Loss of Heat through Furnace Walls, W. T. Ray, Henry Kresinger; The Panama Railroad, Ralph Budd.

MEETINGS TO BE HELD IN THE ENGINEERING BUILDING

Date	Society	Secretary	Time
October			
15	New York Railroad Club	H. D. Vought	8.15
19	New York Telephone Society	T. H. Lawrence	8.00
27	Municipal Engineers of City of New York	C. D. Pollock	8.15
November			
3	Wireless Institute	S. L. Williams	7.30
4	Blue Room Engineering Society	W. D. Sprague	8.00
5	Explorers' Club	H. C. Walsh	8.30
6	Amer. Soc. Hungarian Engrs. and Archts	Z. deNemeth	8.30
9	The American Society of Mech. Engrs	Calvin W Rice	8.00
11	Illuminating Engineering Society	P. S. Millar	8.00
12	American Institute of Electrical Engrs	R. W. Pope	8.00
16	New York Telephone Society	T. H. Lawrence	8.00
17–18	Empire State Gas and Electric Asso	C. H. B. Chapin	All day
18–19	Naval Architects and Marine Engrs	W. H. Baxter	All day
19	New York Railroad Club	H. D. Vought	8.15
24	Municipal Engineers of City of New York	C. D. Pollock	8.15

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[1]Deceased.

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1909

On a Standard Tonnage Basis for Refrigeration

D. S. JACOBUS — G. T. VOORHEES
A. P. TRAUTWEIN — PHILIP DE C. BALL
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On Society History

JOHN E. SWEET — H. H. SUPLEE
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On Constitution and By-Laws

CHAS. WALLACE HUNT, *Chairman* — F. R. HUTTON
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On Conservation of Natural Resources

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On International Standard for Pipe Threads

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On Thurston Memorial

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On Hudson-Fulton Celebration

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On Standards for Involute Gears

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On Power Tests

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On Land and Building Fund

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1909

On John Fritz Medal

HENRY R. TOWNE (1) — F. R. HUTTON (3)
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On Board of Trustees United Engineering Societies Building

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J. W. LIEB, JR., CHAIRMAN OF THE LIBRARY COMMITTEE OF THE AM. SOC. M. E.

On National Fire Protection Association

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On Joint Committee on Engineering Education

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On Government Advisory Board on Fuels and Structural Materials

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